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STEAM CAR CONTROL ANALYSIS

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Southfield, Michigan

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SECTION 1

SUMMARY

During the quarterly reporting period a program redirection was initiated. Although details of the redirection have not been finalized, the essential parts of the redirected effort are shown in the revised program plan of Figure 1-1. The old Task 5 "Control Cost Estimates," is replaced with a new Task 5, "Facsimile Control Hardware." The new Task 5 includes three key efforts: 1) system integration, 2) feedwater control facsimile hardware (bypass valve), 3) facsimile damper combustion air controller and fuel/air ratio controller. An additional task change is the postponement of the final reporting date to the end of July 1972.

During the quarterly reporting period, the design of the SES Model 5 Steam Generator was changed. In addition to some minor adjustment in metal weights and heat transfer coefficients in the various parts of the steam generator, the two significant changes were the addition of a fifth bare-tube superheater pass in the steam generator and the rearrangement of the combustion gas flow path. With the new path the bare-tube superheater pass is located next to the burner. The hybrid computer simulation was modified during the quarterly period to reflect these changes. A revision of the hybrid computer program is given in detail in Section 2. In addition to the changes resulting from the new steam generator design, the hybrid computer program simulation was modified to include the simulation of engine-driven auxiliaries such as feedwater pump and combustion air blower through a common variable speed drive. The control of this drive was evaluated with the inclusion of evaluation of other design restrictions, such as the need for a minimum combustion air blower speed and also the accommodation of other limiting factors in combustion air blower and feed pump speeds.

Typical hybrid computer results are shown to illustrate the realistic behavior of the computer model. Examples of evaluation of various control concepts and elaborate interactions between choices made for various individual components are also given in Section 3. An evaluation of the effects of specific design details, such as blower characteristics and various ways of mechanizing the pump control concepts, are presently in progress. Results, to date, indicate that the most severe demand on the pressure and temperature control system occurs at the extremely low steam and combustion gas flow rates occurring during the idle operating condition.

A digital computer simulation of the burner and combustor system was established during the reporting period to provide the flow-pressure map of the burner system when operated using an engine-driven combustion air blower. The results of this computer program were used to describe the burner system in the hybrid computer simulation.

A pressure operated feedwater bypass valve was also designed during the quarterly period. The design of the valve is such that direct steam operation of the bypass valve is possible for the evaluation of the simplest trim control mode. However, the design is flexible enough to accommodate the addition of an external operator, if required, to evaluate experimentally more elaborate bypass valve control concepts.

The adaptation of a commercially available Bendix RSA fuel/air ratio control unit was initiated during the quarterly reporting period. The use of this unit is planned for the facsimile testing. This closed-loop fuel/air ratio unit will offer maximum flexibility for the optimization of the burner emission characteristics during the facsimile burner system testing.

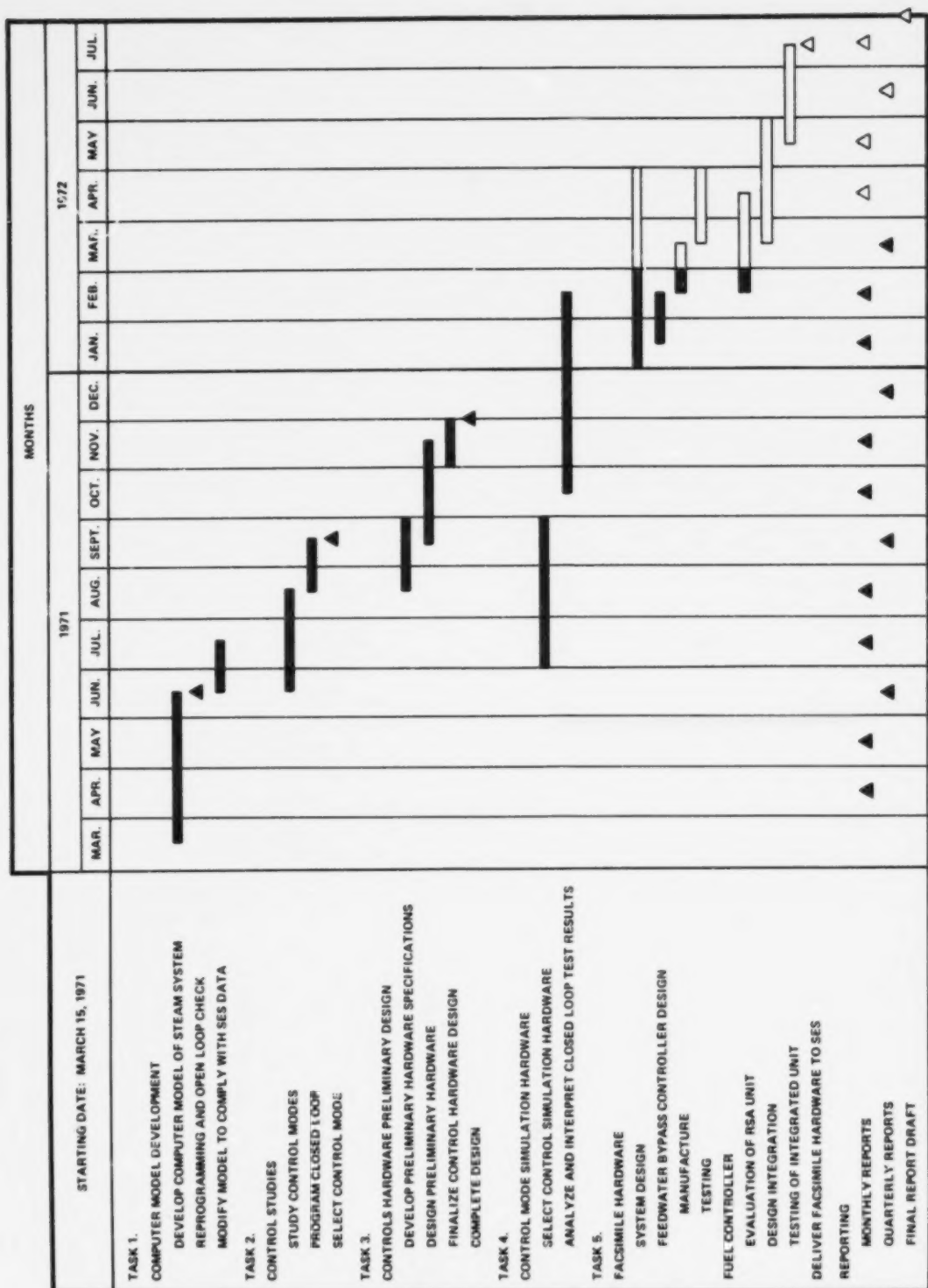


Figure 1-1 - Program Plan

SECTION 2

HYBRID COMPUTER SIMULATION

The following section revises the hybrid computer model presented in the May 1971 monthly report. The steam generator model analytical results have been compared with SES-4 open-loop boiler results. The only significant revision to the model was the replacement of the enthalpy first order time constants with a digital representation of the thermal wave propagation. Additional changes were made to revise the coefficients to correspond with the new SES-5 steam generator sizing. The model, as presented, describes the present hybrid simulation and it includes, in addition to the steam generator, the expander, the vehicle, the auxiliaries and the current control concept. Closed-loop control evaluations are in progress using the model.

2.1 STEAM GENERATOR ANALYTICAL MODEL

All variables used in the analytical models are described in alphabetical order by symbol in the nomenclature at the end of this section. In addition to a short description of the variable itself, the specific units required to be compatible with the other variables are listed. Also, when applicable, typical values of the variables describing the SES-5 steam generator are entered. It must be noted, however, that these values described current design and, at a later time, some of the coefficients may be revised to reflect the final SES design.

Burner

The burner model assumes constant air-fuel ratio:

$$W_g = W_F (1 + A/F) \quad (1)$$

$$T_g = \text{constant} \quad (2)$$

Gas residence time in the burner and boiler is assumed to be negligible compared to the boiler dynamics. In a later section, gas flow prediction using engine driven auxiliaries will be described.

Feed Water Pump

Feed pump characteristics may be elaborated during the program. For the present, description of a controllable, variable displacement pump may be sufficient. A more detailed analytical description of a

variable-ratio engine driven fixed displacement pump is given in the control evaluation section.

$$T_{fei} = \text{constant} \quad (3)$$

$$W_{fei} = D_p \rho_{fei} \dot{\theta}_p \quad (4)$$

Other feedwater supply system characteristics can be readily incorporated in the analysis.

Steam Generator

The steam generator model is subdivided into three fixed length sections: economizer, boiler and superheater sections. The information flow diagram representing the steam generator is shown in Figure 2-1.

Economizer

The flow through the economizer is nearly incompressible.

$$W_{fei} = W_{feo} = W_{fe} \quad (5)$$

The energy equation for the economizer section yields

$$m_{fe} \frac{d}{dt} h_e = Q_{mfe} + W_{fe} (h_{ei} - h_{eo}) \quad (6)$$

Pressures losses through the length of the economizer section are described in terms of viscous flow losses.

$$P_{ei} - P_{eo} = \frac{2 f_e L_e}{\rho_{fei} g_o D_e} \frac{W_e^2}{A_{xe}^2} \quad (7)$$

The metal-to-fluid heat flow is a function of the temperature difference between the metal and the fluid.

$$Q_{mfe} = u_{fe} A_{Hfe} (T_{me} - T_{fe}) \quad (8)$$

$$u_{fe} = C_{le} \left(\frac{W_{fe}}{W_{fe*}} \right)^{0.8} \quad (9)$$

The enthalpy of the inlet fluid is a nearly linear function of the fluid inlet temperature.

$$h_{ei} = C_{fe} T_{fei} + h_{oe} \quad (10)$$

The gas-to-metal heat flow is based on the temperature difference between the inlet gas to the economizer and the metal temperature at the inlet. However, for the particular heat exchanger design, the inlet metal temperature is sufficiently close to the fluid temperature at the inlet so that the fluid inlet temperature T_{fei} may be substituted for the heat transfer calculations.

$$T_{gei} - T_{geo} = \epsilon_e (T_{gei} - T_{fei}) = \Delta T_{ge} \quad (11)$$

$$\epsilon_e = \frac{1 - e^{-\frac{N T U_e}{2}}}{1 - \frac{1}{2} e^{-\frac{N T U_e}{2}}} \quad (12)$$

$$N T U_e = 1.1 \left(\frac{W_g}{W_{ge*}} \right) \quad (13)$$

For computation simplicity the above coefficients may be included into a single function describing the gas-to-metal heat conductance, f_e .

$$f_e (W_g) = C_{ge} \epsilon_e \quad (14)$$

$$W_g \Delta h_{ge} = C_{ge} (\Delta T_{ge}) W_g = Q_{gme} = f_e (W_g) (T_{gei} - T_{fei}) W_g \quad (15)$$

Metal temperature calculation is based on the difference between the heat flow from gas to metal and metal to fluid in addition to the mass and specific heat of the metal.

$$m_{me} C_{me} \frac{d}{dt} T_{me} = Q_{gme} - Q_{mfe} \quad (16)$$

The fluid output temperature at the economizer outlet can be computed on the basis of outlet enthalpy.

$$T_{feo} = \frac{1}{C_{fe}} (h_{eo} - h_{oe}) \quad (17)$$

The model presented so far accommodates static heat transfer characteristics and primary dynamics of the economizer section. The distributed nature of the thermal wave is not sufficiently represented for the economizer section. An approximate description of the thermal wave propagation time was used.

$$\tau_{eh} = \frac{m_{me} C_{me}}{W_{fe} C_{fe}} \quad (18)$$

Equation (18) indicates that the thermal wave propagation time is a function of fluid flow. During the anticipated operating range of the steam generator, the changes in thermal wave propagation must be accommodated. The most convenient inclusion of the variable thermal wave propagation time was through the use of the digital portion of the hybrid computer. The outlet enthalpy of the economizer was thus delayed prior to use in the boiler energy equation.

$$h_{feo}^* = h_{feo} \frac{1 - e^{-\tau_{eh}s}}{\tau_{eh}s} \quad (19)$$

Inclusion of the thermal wave propagation time reuses the metal thermal inertia in addition to the account for m_{me} in equation (16). The repeated use of the metal mass resulted in a slower computer result than the experimental results indicated. Reducing the metal mass to $0.75 m_{me}$ in equation (16) offered a simple empirical solution to this problem. The same correction was also made in the boiler and superheater sections of the steam generator model.

Boiling Section

Conservation of mass in the fixed length representation of the boiling section may be stated as

$$V_b \frac{d}{dt} \rho_{fb} = W_{feo} - W_{fbo} \quad (20)$$

The energy equation for the boiling section defines the time rate of change of average fluid density and internal energy for the volume of the boiling section.

$$V_b \frac{d}{dt} (\rho_{fb} U_{fb}) = Q_{mfb} + W_{feo} h_{feo}^* - W_{fbo} h_{fbo} = V_b (\dot{\rho}_{fb} U_{fb} + \dot{U}_{fb} \rho_{fb}) \quad (21)$$

Thus the rate of change of internal energy in the boiling section is defined as

$$\dot{U}_{fb} = \frac{1}{\rho_{fb} V_b} (Q_{mfb} + W_{feo} h_{feo}^* - W_{fbo} h_{fbo} - \dot{\rho}_{fb} U_{fb} V_b) \quad (22)$$

The outlet flow from the boiler section is based on viscous flow calculations for the given length of pipe and the square root of pressure difference between boiler and superheater pressures.

$$W_{fbo} = \left(\frac{\rho_{fb}^* g_o D_b}{2 f_b L_b} \right)^{1/2} A_{xb} \sqrt{P_b - P_s} \quad (23)$$

The gas-to-metal heat flow rate is again defined as a function of temperature difference between the gas inlet temperature and the boiler metal temperature.

$$Q_{gmb} = W_g C_{gb} (T_{gbi} - T_{mb}) \left(1 - e^{-NTU_b \left(\frac{W_g}{W_{gb}^*} \right)^{-0.4}} \right) \quad (24)$$

This equation, again, may be simplified by representing the heat conductance by a single function, f_b .

$$Q_{gmb} = f_b (W_g) (T_{gbi} - T_{mb}) W_g \quad (25)$$

With f_b defined as follows:

$$f_b (W_g) = C_{gb} (1 - e^{-x}) \quad (26)$$

$$x = NTU_b \left(\frac{W_g}{W_{gb}^*} \right)^{-0.4} \quad (27)$$

The outlet gas temperature is calculated from the inlet gas temperature and the temperature loss due to the heat flow to the metal.

$$T_{gbo} = T_{gbi} - \frac{Q_{gmb}}{W_g C_{gb}} \quad (28)$$

The metal temperature time derivative is again related to the gas-to-metal and metal-to-fluid heat flow rate and the mass and specific heat of the metal of this section.

$$m_{mb} C_{mb} \frac{d}{dt} T_{mb} = Q_{gmb} - Q_{mfb} \quad (29)$$

Heat flow rate from the metal to the fluid in the boiler section is again governed by the temperature difference between the metal and the fluid of the boiler. The coefficient of heat conductance in this particular section may be assumed as a large constant, U_{fb} , due to the high heat transfer rate in the boiling section.

$$Q_{mfb} = u_{fb} A_{Hfb} (T_{mb} - T_{fb}) \quad (30)$$

The basic definition of enthalpy, as the sum of internal energy plus a PV term, is used to calculate enthalpy of the boiler fluid.

$$h_{fb} = U_{fb} + \frac{P_b}{J \rho_{fb}} \quad (31)$$

The definition of specific volume and specific density of the fluid is used to calculate the specific volume of the fluid.

$$V_{fb} = 1/\rho_{fb} \quad (32)$$

A simplified set of equations are used to describe the equation of state for the fluid in the boiling section. This simplified relationship was found to be satisfactory to evaluate the control characteristics of the boiler. Comparison with the experimental data obtained from the SES-4 boiler was used to establish the validity of this assumption. The pressure in the boiler is described as a function of the temperature.

$$P_b = A + B T_{fb} + \frac{C}{2} T_{fb}^2 \quad (33)$$

On the basis of the Clapeyron relationship for the boiler

$$T_{fb} \frac{dP_b}{dT_{fb}} = \frac{(U_{fb} - U_L) J}{V_{fb} - V_L} + P_b \quad (34)$$

The internal energy of the liquid is a linear function of temperature.

$$U_L = D T_{fb} - E \quad (35)$$

Using the Clapeyron relationship, the temperature of the fluid in the boiler can be defined as follows:

$$T_{fb} = (-b + \sqrt{b^2 - 4ac})/2a \quad (36)$$

$$a = \frac{C}{2} (v_{fb} - v_L) \quad (37)$$

$$b = 460C (v_{fb} - v_L) + DJ = 920a + DJ \quad (38)$$

$$c = -J (E + U_{fb}) - (v_{fb} - v_L) (A - 460 B) \quad (39)$$

The thermal wave propagation time is again included to improve the model's representation of the distributed thermal capacitance of the monotube steam generator.

$$\tau_{bh} = \frac{m_{mb} C_{mb}}{W_{fbo} C_{fb}} \quad (40)$$

The resulting thermal delay is again represented as a variable distributed delay time as a function of boiler outlet flow

$$h_{fbo}^* = h_{fbo} \frac{1 - e^{-\tau_{bh} s}}{\tau_{bh} s} \quad (41)$$

Superheater

Conservation of mass defines the change of density in the fixed section.

$$V_s \frac{d}{dt} \rho_{fs} = W_{fbo} - W_s \quad (42)$$

The energy equation in the superheater may be simplified because the contribution through the rate of change, ρU , is negligible.

$$V_s \frac{d}{dt} (\rho_{fs} U_{fs}) = Q_{mfs} + W_{fbo} h_{fbo}^* - W_s h_s = 0 \quad (43)$$

The superheater pressure loss is again calculated on the basis of viscous flow and the time constant relating the volumetric charge time of the superheater volume.

$$P_s = P_{fs} - \frac{2 f_s L_s W_s^2}{\rho_s^* g_o D_s A_{xs}^2} \left(\frac{1}{\tau_{sp} S + 1} \right) \quad (44)$$

The heat flow rate from the gas to metal again is defined as a function of gas flow rate and temperature difference between the superheater inlet gas temperature and metal temperature.

$$Q_{gms} = W_g C_{gs} (T_{gsi} - T_{gso}) = W_g C_{gs} \epsilon_s (T_{gsi} - T_{ms}) \quad (45)$$

This equation may be simplified as follows

$$Q_{gms} = f_s (W_g) (T_{gsi} - T_{ms}) W_g \quad (46)$$

f_s flow function in the superheater section is defined as follows

$$f_s (W_g) = C_{gs} \epsilon_s \quad (47)$$

$$\epsilon_s = 0.7 \left(1 - 0.96 e^{-NTU_s \left(\frac{W_g}{W_{gs}^*} \right)^{-0.4}} \right) \quad (48)$$

Outlet temperature of the gas leaving the superheater is calculated on the basis of heat loss.

$$T_{gso} = T_{gsi} - \frac{Q_{gms}}{W_g C_{gs}} \quad (49)$$

The metal-to-fluid heat flow rate is again calculated on the basis of temperature difference.

$$Q_{mfs} = u_{fs} A_{Hfs} (T_{ms} - T_s) \quad (50)$$

The heat transfer rate is defined as a 0.8 power of the fluid flow rate.

$$u_{fs} = C_{1s} \left(\frac{W_s}{W_s^*} \right)^{0.8} \quad (51)$$

The metal temperature is calculated on the basis of heat flow difference between the gas-to-metal and metal-to-fluid flow rates. In addition, a term describing the equivalent mass of the superheater steam and its specific heat is included in the equation. This correction, however, is small.

$$(m_{fs} C_{ps} + m_{ms} C_{ms}) \frac{d}{dt} T_{ms} = Q_{gms} - Q_{mfs} \quad (52)$$

The temperature of the working fluid in the superheater section is calculated on the basis of enthalpy and specific heat.

$$T_s = (h_s - h_{os}) / C_{ps} \quad (53)$$

The superheater pressure is a nearly linear function of the superheater temperature.

$$P_{fs} = C_{Tps} T_{fs} \rho_{fs} \quad (54)$$

The superheater outlet temperature, T_s' , may be solved by substituting the expression for Q_{mfs} , from equation (50), and the enthalpy, h_s , from equation (53), into the energy balance equation (43).

$$(C_{ps} T_s' + h_{os} - h_{fbo}^*) W_s = W_{fbo} h_{fbo}^* + U_{fs} A_{Hfs} (T_{ms} - T_s') \quad (54)$$

Equation (54) now may be solved for the superheater outlet temperature

$$T_s' = \frac{1}{U_{fs} A_{Hfs} + W_s C_{ps}} (U_{fs} A_{Hfs} T_{ms} + W_{fbo} h_{fbo}^* - W_s h_{os}) \quad (56)$$

The simplified equation of state used in the superheater section allows the insertion of the distributed effect of thermal propagation in series with the superheater outlet temperature.

$$T_s = T_s' \frac{1 - e^{-\tau_{sh} s}}{\tau_{sh} s} \quad (57)$$

and similar to the economizer and boiler section

$$\tau_{sh} = \frac{m_{ms} C_{ms}}{W_s C_{fs}} \quad (58)$$

The steam flow is normally calculated in the expander section, however, a choked flow variable area throttle is used to compare steam generator operation independent from the expander to verify the analytical model with the experimental results. For the comparison, steam flow was calculated on the basis of throttle admittance and steam conditions.

$$W_s = K_E \frac{P_s}{\sqrt{T_s + 460}} \quad (59)$$

2.2 EXPANDER AND VEHICLE

For the purposes of control system evaluation, the expander may be characterized as a continuous flow device. This assumption is valid because the steam generator dynamics are much slower than the individual

flow pulsations that result from the opening and the closing of the intake valves on the four cylinder expander. Engine output torque and mass flow consumed by an expansion device can be accurately predicted using the analytical model described by Taplin and Gregory*. Using this analytical model, the engine torque output in general is defined by a torque coefficient D_{MT} .

$$\text{Torque} = D_{MT} \times P \quad (60)$$

The flow consumed by the expander is defined by a volumetric coefficient D_{MV} .

$$\text{Weight flow} = D_{MV} \rho_s \dot{\theta} \quad (61)$$

In a variable admission device, such as the SES expander, both D_{MT} and D_{MV} are also a function of the admission angle and engine RPM. It will be shown that a modified continuous flow model will simply and accurately describe the SES expander on the basis of available flow and indicated horsepower data. The information diagram for the combined expander and load dynamics and engine auxiliaries is shown in Figure 2-2.

The engine torque, T_E , is calculated on the basis of the D_{MT} model,

$$T_E = D_{MT} (P_s - P_c) \quad (62)$$

where D_{MT} is a function of both engine speed and valve position

$$D_{MT} = D_{MT}(\alpha) D(\dot{\theta}) \quad (63)$$

Similarly, the flow demand of the engine is computed

$$W = D_{MV} \frac{P_s}{(T_s + 460) R_s} \dot{\theta} \quad (64)$$

* L. B. Taplin and A. J. Gregory, "Rotary Pneumatic Actuators," Control Engineering, December 1963.

$$D_{MV} = D_{MV}(\alpha) D(\dot{\theta}) \quad (65)$$

The vehicle and accessory loads can be combined into a single second order representation for preliminary control studies. A more elaborate model may be included at a future time as engine performance and auxiliary loads become more firmly defined.

$$T_L = K_1 \ddot{\theta} + K_2 \dot{\theta} + F \quad (66)$$

The vehicle and engine inertia are combined as reflected at the engine RPM.

$$\ddot{\theta} = (T_E - T_L) \frac{1}{I_E + I_L} \quad (67)$$

2.3 AUXILIARIES AND FLOW CONTROL MECHANIZATION

The present simulation describes an engine driven combustion air blower and feedpump operated through a variable ratio transmission. The transmission ratio is controlled by a central controller. A simple steam-pressure-operated bypass valve with controlled slew rate is used to implement a closed-loop pressure control. The closed-loop temperature controller actuates an air damper valve to regulate the combustion gas flow. The separate fuel controller is assumed to maintain fuel-air ratio, and is not included in the present system simulation.

The pump speed is controlled by the variable speed transmission, represented as a second order system.

$$\dot{\theta}_p = R_p \dot{\theta} \quad (68)$$

$$R_p = \frac{\frac{D_{MV}}{s^2} + \frac{R_s T_s^* \rho_{fe1} D_p}{\frac{\omega_{Rp}^2}{2} + \frac{2\phi_{ms}}{\omega_{Rp}} s + 1}}{P_s^*} \quad (69)$$

Blower speed is directly proportional to the pump speed.

$$\dot{\theta}_B = R_B \dot{\theta}_P \quad (70)$$

Pump delivery is defined by pump speed

$$W_P = D_P Q_{fei} \dot{\theta}_P \quad (71)$$

Steam generator inlet flow is defined by conservation of mass

$$W_{fe} = W_P - W_R \quad (72)$$

The bypass flow is calculated on the basis of pressure, fluid properties and valve opening

$$W_R = A_R \sqrt{2 g_o \rho_{fei} P_{fei}} \quad (73)$$

The valve opening is defined by the geometry of the design and instantaneous valve position

$$A_R = K_{B2} x^2 \quad (74)$$

Valve position calculation is based on the solution of a second order system

$$\ddot{x} = \frac{\Delta F}{m_{Dp}} \quad (75)$$

$$\Delta F = P_s A_\Delta - b_o \dot{x} - F_p - K_s x \quad (76)$$

The combustion gas flow is defined by blower speed, blower characteristics and air damper opening

$$W_g = W_g^* (\dot{\theta}_B) \frac{A_D}{A_{D \max}} \quad (77)$$

where blower opening is controlled by the temperature error signal. The rapid time response of the damper system may be neglected for the initial evaluation

$$\frac{A_D}{A_{\max}} = K_A (T_{\text{set}} - T_s) \quad (78)$$

2.4 SES-5 STEAM GENERATOR AND EXPANDER REPRESENTATION

In order to increase the solution speed of the overall hybrid system simulation, several relationships were precomputed and the results of these calculations were used to program analog nonlinear function generators. In the following, results of these calculations, as applicable to the SES-5 steam generator, are presented.

The gas-to-metal heat transfer equations (15), (25), and (46) require the functions $f_e(W_g)$, $f_b(W_g)$, and $f_s(W_g)$ as a function of gas flow, W_g . Figure 2-3 presents this information in the form currently used in the hybrid simulation.

The metal-to-fluid heat transfer coefficients, U_e and U_b , as a function of fluid flow, W_f , are shown in Figure 2-4. The functional relations are used in equations (3) and (50) to calculate metal-to-fluid heat flow in the economizer and the superheater. In the boiling section, a fixed heat transfer coefficient is used.

The expander DM_T and DM_V curves were generated on the basis of SES data describing expander performance at various operating conditions defined by valve angles from 20 to 70 degrees at 500, 1500, and 2500 RPM engine speeds. The 500 RPM data set was used to represent both the torque and volumetric coefficient maps of the engine in the hybrid simulation. Results are given in Figure 2-5. The speed correction curve, as used in equations (63) and (65), is given in Figure 2-6.

Engine losses, auxiliary power consumption, and vehicle load were combined into a single load torque curve for the initial studies. A second order least squares fit to this curve was used to represent loads to the engine in high or low gear. The following expressions were used to calculate total load.

$$\text{Load} = \text{Engine Friction} + \text{Auxiliaries} + \text{Road Load}$$

$$\text{Auxiliaries} = \text{Feedpump} + \text{Burner} + \text{Fan Power}$$

The engine friction horsepower load was defined as

$$\text{Engine Friction} = 4 \times 4.41 \times 10^{-5} \text{ RPM } (11 + \text{RPM}/100)$$

Auxiliary horsepower loads were defined as

$$\text{Feedpump Power} = 2.42 \times 10^{-3} \times \text{Steam Flow (lb/hr)}$$

$$\text{Burner Power} = 0.68 + 1.18 (\text{Steam flow (lb/hr)}/1290)^3$$

$$\text{Fan Power} = 2.084 - 5.94 \times 10^{-3} \text{ RPM} + 4.36 \times 10^{-6} \text{ RPM}^2$$

Road Load was defined in terms of grade and vehicle speed for a 4600 pound vehicle.

$$\begin{aligned} \text{Road Load} = V \text{ (ft/sec)}/550 & \left[4600/\sqrt{1 + G^2} (\%) (G(\%) + 0.0154) + \right. \\ & \left. + 9.91 \times 10^{-2} V(\text{ft/sec}) + 1.44 \times 10^{-2} V^2(\text{ft/sec}) \right] \end{aligned}$$

The conversion factors between engine RPM and vehicle speed were given as

$$\text{RPM} = 20.50 V(\text{ft/sec}) \quad \text{Second Gear}$$

$$\text{RPM} = 13.16 V(\text{ft/sec}) \quad \text{High Gear}$$

The composite load torque is shown as a function of engine speed for the high-gear condition in Figure 2-7, and for the second-gear engagement in Figure 2-8. Least squares fits of both curves are also shown in Figures 2-9 and 2-10 respectively. The least squares fit representing the high-gear conditions is used at the present for the load torque in the current control system studies as required by equation (66).

Both the maximum and minimum value of the cut-off valve were limited as a function of engine speed as shown in Figure 2-11. The maximum limit is required to hold the expander steam consumption within the maximum output of the steam generator. The minimum setting provides the idle speed regulation for the expander.

The vehicle ratio drive settings are similarly controlled to avoid over-speeding of the auxiliaries are shown in Figure 2-12. Maximum available ratio from the drive is 0.9 and the minimum is 0.3 for the design currently considered.

NOMENCLATURE

<u>Symbol</u>	<u>Units</u>	<u>Value</u>	<u>Description</u>
a	1/°F		Equation-of-state constant
A	lb/in ²	4267.5	Equation-of-state coefficients, boiler
A _D	in ²		Air damper opening
A _{D max}	in ²		Maximum damper opening
A/F			Air-Fuel ratio
A _{Hfe}	in ²	625	Heat transfer area, fluid side, economizer
A _{Hfb}	in ²	1150	Heat transfer area, fluid side, boiler
A _{Hfs}	in ²	373	Heat transfer area, fluid side, superheater
A _R	in ²	3.518 x 10 ⁻³	Bypass flow area
A _Δ	in ²	0.169	Bypass Differential Area
A _T	in ²	0.05373	Flow area, throttle
A _{xb}	in ²	0.192	Flow area, boiler tube
A _{xe}	in ²	0.107	Flow area, economizer tubes
A _{xs}	in ²	0.192	Flow area, superheater
s			Equation-of-state constant
b _o	lb sec/in	0.444	Bypass Damping Coefficient
B	lb/in-°F	-20.35	Equation-of-state coefficients, boiler
c	°F		Equation-of-state constant

<u>Symbol</u>	<u>Units</u>	<u>Value</u>	<u>Description</u>
C	lb/in ² -°F ²	0.0527	Equation-of-state coefficients, boiler
C ₁	√°R/sec	0.4049	Superheated steam, gas coefficient
C _{1e}	btu/(in ² -sec-°F)	3.95 x 10 ⁻³	Film coefficient, economizer, fluid side
C _{1s}	btu/(in ² -sec-°F)	1.49 x 10 ⁻³	Film coefficient, superheater, fluid side
C _d		0.80	Orifice coefficient, throttle
C _{fe}	btu/lb-°F	1.09	Specific heat, fluid, economizer
C _{gb}	btu/lb-°F	0.3263	Specific heat, gas, boiler
C _{ge}	btu/lb-°F	0.2932	Specific heat, gas, economizer
C _{gs}	btu/lb-°F	0.3554	Specific heat, gas, superheater
C _{mb}	btu/lb-°F	0.11	Specific heat, metal, boiler
C _{me}	btu/lb-°F	0.11	Specific heat, metal, economizer
C _{ms}	btu/lb-°F	0.11	Specific heat, metal, superheater
C _{ps}	btu/lb-°F	0.697	Specific heat, fluid, superheater
C _{Tps}	in/°F	1437	Superheater conversion coefficient, temperature pressure
D	btu/lb-°F	1.26	Equation-of-state coefficients, boiler
D _b	in.	0.495	Boiler tube I.D.
D _e	in.	0.37	Diameter, economizer fluid tube
D _{MT} (α)	in ³ -sec/rad		Expander torque constant

<u>Symbol</u>	<u>Units</u>	<u>Value</u>	<u>Description</u>
$D_{MV}(\alpha)$	$\text{in}^3\text{-sec/rad}$		Expander volumetric constant
D_p	in^3/rad	0.0872	Water pump displacement coefficient
D_s	in.	0.495	Tube diameter, superheater
$D(\dot{\theta})$			Velocity coefficient, expander constants
E	btu/lb	148	Liquid, internal energy equation coefficient
F	in-lb	1399.2	Vehicle rolling resistance revised
f_b		4×10^{-3}	Friction factor, boiler
$f_b(W_g)$			Boiler heat transfer function
f_e		4×10^{-3}	Friction factor, economizer
$f_e(W_g)$			Economizer gas-to-metal heat transfer function
$f_l\left(\frac{P_M}{P_S}\right)$			Flow function
F_p	lb	160.55	Bypass spring preload
f_s		4×10^{-3}	Friction factor, superheater
$f_s(W_g)$			Superheater heat transfer function
g_o	in/sec^2	386	Gravitational constant for mass conversion
h_e	btu/lb		Enthalpy - average economizer
h_{ei}	btu/lb	190.4	Enthalpy - economizer, inlet
h_{eo}	btu/lb		Enthalpy - economizer, outlet
h_{fb}	btu/lb		Enthalpy - fluid, boiler
h_{fbo}	btu/lb		Enthalpy - fluid, boiler outlet

<u>Symbol</u>	<u>Units</u>	<u>Value</u>	<u>Description</u>
h_{oe}	btu/lb	-51	Enthalpy - baseline, economizer
h_{os}	btu/lb	864.35	Enthalpy - baseline, superheater
h_s	btu/lb		Enthalpy, superheater output
h_{fsi}	btu/lb	1187.8	Enthalpy, superheater inlet
h_v	btu/lb		Enthalpy, vapor
I_E	in-lb-sec ² /rad	8	Engine inertia
I_L	in-lb-sec ² /rad	901 371	Reflected load inertia - High Gear - Second Gear
J	in-lb/btu	9335	Thermal conversion coefficient
k		1.273	Ratio of specific heats, superheated steam
K_A	1/°F	3.5×10^{-3}	Damper positioner gain
K_E	in ² √°F/sec	0.3605	Simulated throttle admittance
k_v			Expander velocity constant
K_s	lb/in	228	Bypass spring constant
K_1	in-lb-sec ² /rad ²	7.626×10^{-2}	Load torque - second order coefficient
K_2	in-lb-sec/rad	-9.0189	Load torque - speed coefficient
L_b	in.	739	Length of boiler
L_e	in.	538	Length of economizer
L_s	in.	240	Length of superheater
m_{DP}	lb-sec ² /in	2.163×10^{-4}	Mass of bypass valve
m_{fb}	lb		Mass of fluid, boiler

<u>Symbol</u>	<u>Unit</u>	<u>Value</u>	<u>Description</u>
m_{fe}	lb	1.7	Mass of fluid, economizer
m_{fs}	lb	4.20×10^{-2}	Mass of fluid, superheater
m_{mb}	lb	39.3×0.75	Mass of metal, boiler
m_{me}	lb	28.4×0.75	Mass of metal, economizer
m_{ms}	lb	7.8×0.75	Mass of metal, superheater
NTU_b		1.10	Heat transfer units, boiler
NTU_e		1.10	Heat transfer units, economizer
NTU_s		0.12	Heat transfer units, superheater
P_b	psia		Boiler pressure
P_c	psia	20	Condenser pressure
P_{DP}	lb/in ²		Damping chamber pressure
P_{ei}	psig		Pressure, economizer inlet
P_{eo}	psia		Pressure, economizer outlet
P_{fs}	psia		Average pressure, superheater
P_M	psia		Steam manifold pressure
P_p	lb/in ²		Pump output pressure
P_s	psia		Superheater outlet pressure
P_s^*	psia	1000	Nominal superheater outlet pressure
Q_{gmb}	btu/sec		Heat flow, gas to metal, boiler
Q_{gme}	btu/sec		Heat flow, gas to metal, economizer
Q_{gms}	btu/sec		Heat flow, gas to metal, superheater

<u>Symbol</u>	<u>Unit</u>	<u>Value</u>	<u>Description</u>
Q_{mfb}	btu/sec		Heat flow, metal to fluid, boiler
Q_{mfe}	btu/sec		Heat flow, metal to fluid, economizer
Q_{mfs}	btu/sec		Heat flow, metal to fluid, superheater
R_B		2.0	Variable speed drive ratio - blower drive
R_P		0.3-0.9	Variable speed drive ratio - pump drive
R_s	in/°R	1027.4	Gas constant, superheated steam
R_v	in/°R	1027.4	Gas constant, vapor, superheater
T_E	in-lb		Torque output, expander
T_{fb}			Average fluid temperature, boiler
T_{fe}	°F		Fluid temperature, economizer, average
T_{fei}	°F	220	Fluid temperature, economizer inlet
T_{feo}	°F		Fluid temperature, economizer outlet
T_{fs}	°F		Average temperature, fluid, superheater
T_{fsi}	°F		Fluid temperature, superheater inlet
T_g	°F	3200	Gas temperature burner output
T_{gbi}	°F		Gas temperature, boiler inlet
T_{gbo}	°F		Gas temperature, boiler outlet

<u>Symbol</u>	<u>Unit</u>	<u>Value</u>	<u>Description</u>
T_{gei}	$^{\circ}F$		Gas temperature, economizer inlet
T_{geo}	$^{\circ}F$		Gas temperature, economizer outlet
T_{gsi}	$^{\circ}F$		Gas temperature, superheater inlet
T_{gso}	$^{\circ}F$		Gas temperature, superheater outlet
T_L	in-lb		Load torque
T_{mb}	$^{\circ}F$		Average metal temperature, boiler
T_{me}	$^{\circ}F$		Metal temperature, economizer, average
T_{ms}	$^{\circ}F$		Average temperature, metal, superheater
T_s	$^{\circ}R$	1460	Superheater outlet temperature
T_s^*	$^{\circ}R$	1460	Nominal superheater outlet temperature
u_{fb}	$btu/(sec-in^2-^{\circ}F)$	1.929×10^{-2}	Heat transfer coefficient, fluid side, boiler
u_{fe}	$btu/(sec-in^2-^{\circ}F)$		Heat transfer coefficient, fluid side, economizer
U_{fs}	btu/lb		Fluid internal energy - superheater
u_{fs}	$btu/sec-in^2-^{\circ}F$		Heat transfer coefficient, fluid side, superheater
u_{gb}	$btu/sec-in^2-^{\circ}F$		Heat transfer units, boiler, gas side
u_{ge}	$btu/sec-in^2-^{\circ}F$		Heat transfer coefficient, gas side, economizer

<u>Symbol</u>	<u>Unit</u>	<u>Value</u>	<u>Description</u>
u_{gs}	btu/sec-in ² -°F		Heat transfer coefficient, gas side, superheater
U_{fb}	btu/lb		Internal energy, fluid, boiler
U_L	btu/lb		Internal energy - liquid, boiler
V_b	in ³	142.1	Volume of boiler
V_{fb}	in ³ /lb		Specific volume, fluid, boiler
V_L	in ³ /lb	37.67	Mean specific volume of liquid, boiler
V_M	in ³	40	Volume, manifold
V_s	in ³	46.1	Superheater volume
W	lb/sec		Fluid flow, expander
W_F	lb/sec	0.0277	Fuel flow rate
W_{fbo}	lb/sec		Fluid flow, boiler outlet
W_{fe}	lb/sec		Fluid flow, economizer
W_{fe}^*	lb/sec	0.333	Nominal fluid flow, superheater
W_{fei}	lb/sec		Fluid flow, economizer inlet
W_{feo}	lb/sec		Fluid flow, economizer outlet
W_{fs}	lb/sec		Average fluid flow rate, superheater
W_{fs}	lb/sec		Average fluid flow rate, superheater
W_{fs}^*	lb/sec	0.333	Nominal fluid flow, superheater
W_g	lb/sec	0.537	Gas flow rate
W_g^*	lb/sec		Blower delivery characteristics

<u>Symbol</u>	<u>Unit</u>	<u>Value</u>	<u>Description</u>
W_{gb}^*	lb/sec	0.537	Reference gas flow rate - boiler
W_{ge}^*	lb/sec	0.537	Reference gas flow rate - economizer (to be set until $T_{fso} = 1000^\circ\text{F}$)
W_{gs}^*	lb/sec	0.537	Reference gas flow rate - superheater
W_p	lb/sec		Pump output flow
W_R	lb/sec		Bypass flow
W_s	lb/sec		Steam flow, superheater outlet
x	in.	0-0.250	Bypass valve travel
α	degrees	0-60	Admission angle
ΔF	lb		Sum of bypass valve forces
Δh_{fe}	btu/lb		Enthalpy difference, fluid economizer
Δh_{ge}	btu/lb		Enthalpy change, economizer gas
ΔP_b	lb/in ²		Pressure drop, boiler
ρ_{fb}	lb/in ³		Average fluid density, boiler
ρ_{fb}^*	lb/in ³	1.389×10^{-3}	Mean density of fluid in boiler
ρ_{fei}	lb/in ³	0.0346	Fluid density, economizer inlet
ρ_{fs}	lb/in ³		Fluid density, superheater
ρ_s^*	lb/in ³	6.944×10^{-4}	Mean density, superheater
θ	rad		Expander rotation

<u>Symbol</u>	<u>Unit</u>	<u>Value</u>	<u>Description</u>
$\dot{\theta}$	rad/sec		Expander speed
$\ddot{\theta}$	rad/sec ²		Expander acceleration
$\dot{\theta}_B$	rad/sec		Blower speed
$\dot{\theta}_p$	rad/sec		Water pump speed
ϵ_b			Heat exchanger effectiveness coefficient, boiler
ϵ_e			Heat exchanger effectiveness coefficient, economizer
ϵ_s			Heat exchanger effectiveness coefficient, superheater
τ_{eh}		8.7	Economic enthalpy time constant at 0.333 lb/sec
τ_{bh}	sec	15.0	Boiler enthalpy time constant at 0.333 lb/sec
τ_{sh}	sec	4.0	Superheater enthalpy time constant, at 0.333 lb/sec
τ_{sp}	sec	0.068	Superheater, pressure buildup time constant
τ_{wb}	sec	0.008	Boiler, flow inertial time constant
ϵ_{RP}		0.707	Variable drive damping ratio
ω_{RP}	rad/sec	3.14	Variable drive frequency response

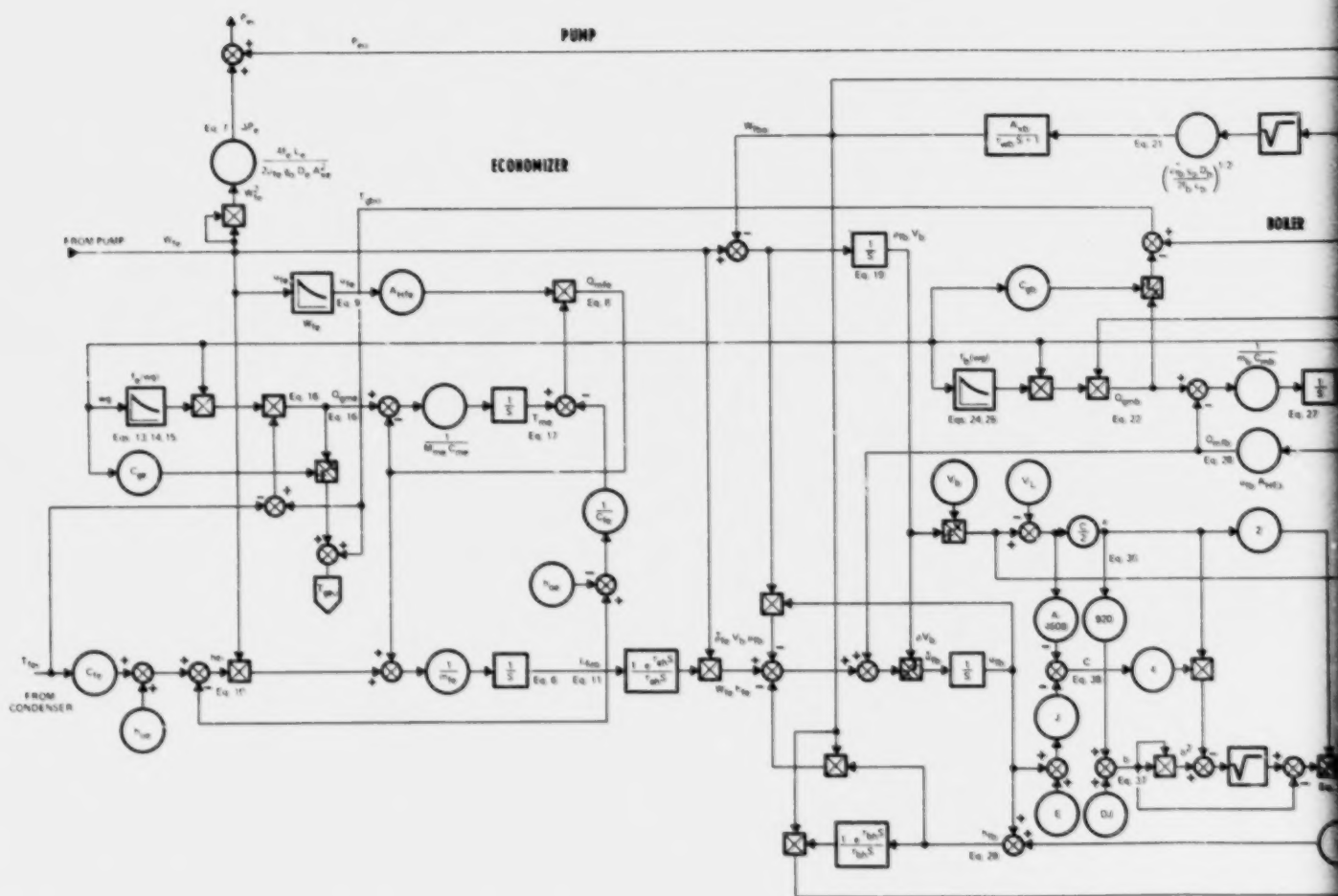


Figure 2-1 - Information Flow Diagram

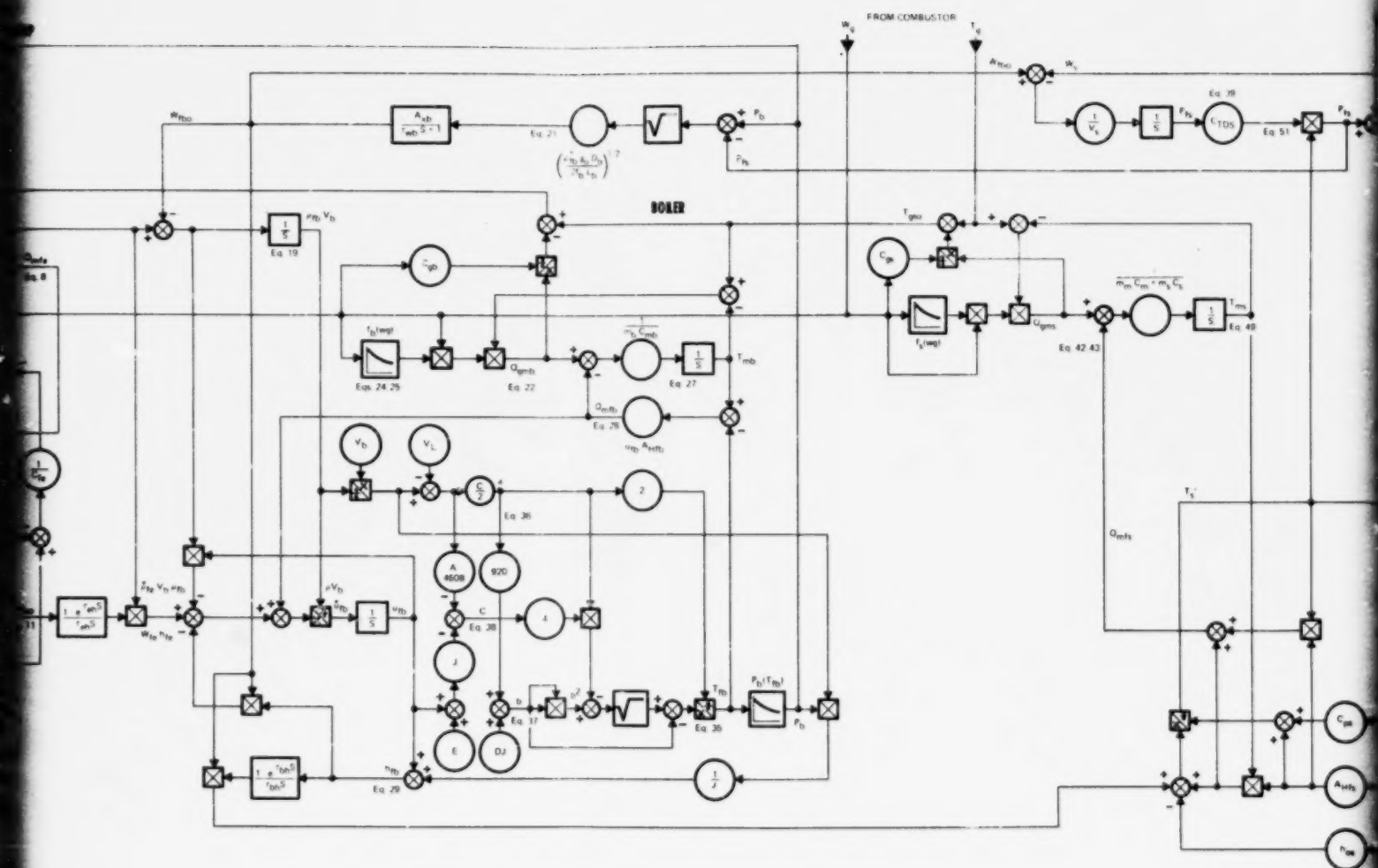


Figure 2-1 - Information Flow Diagram - Steam Generator



2-27

HX-GAS 11:01 02/04/72 FRI.

WGB= .537

INPUT 4 SCALE FACTORS
1,1,1,1

WG	FGE	FGB	FGS	FGE/WG	FGB/WG	FGS/WG
.0268	.007181	.008533	.002461	.2674	.3178	.0917
.0537	.013483	.016417	.004070	.2511	.3057	.0758
.0805	.019303	.023775	.005461	.2396	.2952	.0676
.1074	.024787	.030727	.006732	.2308	.2861	.0627
.1342	.030014	.037353	.007923	.2236	.2782	.0590
.1611	.035034	.043707	.009056	.2175	.2713	.0562
.1879	.039880	.049830	.010142	.2122	.2651	.0540
.2148	.044576	.055753	.011191	.2075	.2596	.0521
.2416	.049142	.061500	.012210	.2034	.2545	.0505
.2685	.053593	.067090	.013203	.1996	.2493	.0492
.2953	.057939	.072540	.014173	.1962	.2456	.0480
.3222	.062192	.077862	.015123	.1930	.2417	.0469
.3490	.066359	.083067	.016056	.1901	.2380	.0460
.3759	.070448	.088165	.016973	.1874	.2345	.0452
.4027	.074464	.093164	.017876	.1849	.2313	.0444
.4296	.078412	.098071	.018766	.1825	.2283	.0437
.4564	.082297	.102893	.019644	.1803	.2254	.0430
.4833	.086124	.107634	.020511	.1782	.2227	.0424
.5101	.089895	.112301	.021368	.1762	.2201	.0419
.5370	.093614	.116896	.022216	.1743	.2177	.0414
.5638	.097283	.121425	.023055	.1725	.2154	.0409
.5907	.100906	.125891	.023887	.1708	.2131	.0404
.6175	.104485	.130297	.024710	.1692	.2110	.0400
.6444	.108021	.134646	.025527	.1676	.2089	.0396
.6712	.111518	.138940	.026336	.1661	.2070	.0392
.6981	.114975	.143183	.027140	.1647	.2051	.0389
.7249	.118396	.147377	.027937	.1633	.2033	.0385
.7518	.121782	.151523	.028728	.1620	.2015	.0382
.7786	.125134	.155624	.029515	.1607	.1999	.0379
.8055	.128454	.159682	.030296	.1595	.1982	.0376
CGE	CGB	CGS				
.2932	.326300	.355400				

Figure 2-3 - Heat Transfer Coefficients for Gas-to-Metal Side

PRTHXF 15:52 02/03/72 THU.

W BASE .3330 CIE .395000E-02 C15 .149000E-02
 INPUT 3 SCALE FACTORS
 ? 1,1,1

WF	UE	US	UE/WF	US/WF
.0166	.000360	.000136	.021595	.008146
.0333	.000626	.000236	.018300	.007092
.0499	.000866	.000327	.017335	.006539
.0666	.001090	.000411	.016366	.006174
.0832	.001303	.000492	.015652	.005904
.0999	.001508	.000569	.015091	.005693
.1165	.001705	.000643	.014633	.005520
.1332	.001896	.000716	.014246	.005374
.1498	.002085	.000787	.013916	.005249
.1665	.002269	.000856	.013626	.005140
.1831	.002448	.000924	.013368	.005043
.1998	.002625	.000990	.013138	.004956
.2164	.002799	.001056	.012929	.004877
.2331	.002969	.001120	.012739	.004805
.2497	.003136	.001184	.012564	.004739
.2664	.003304	.001246	.012403	.004679
.2830	.003468	.001308	.012254	.004622
.2997	.003631	.001370	.012114	.004570
.3163	.003791	.001430	.011984	.004521
.3330	.003950	.001490	.011862	.004474
.3496	.004107	.001549	.011747	.004431
.3663	.004263	.001608	.011638	.004390
.3829	.004417	.001666	.011535	.004351
.3996	.004570	.001724	.011437	.004314
.4162	.004722	.001781	.011344	.004279
.4329	.004872	.001838	.011255	.004246
.4495	.005022	.001894	.011171	.004214
.4662	.005170	.001950	.011090	.004183
.4828	.005317	.002006	.011012	.004154
.4995	.005463	.002061	.010936	.004126

INPUT 3 SCALE FACTORS
 ? .333, .00395, .00149

Figure 2-4 - Heat Transfer Coefficients for Metal-to-Fluid Side

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PRTEXP 9:39 11/15/71 M3N.

INPUT FOR 500,1500,2500,XX,XX, RPMS

5 DMT FACTORS

? 6.7878,5.8597,5.1582,1,1

5 DMV FACTORS

? 3.3607,2.8611,2.4893,1,1

TS 1460.00 RS 1027.20 PS 1000.00 DP 1000.00
HPF 30.7493 TF 63025.358 DMVF 238.6666

α 500.00		RPM						
LEAD	IHP	T	DMT	DMV	WS	DMT/DMV	NDMT	NDMV
20.	10.13	1277.	1.2769	.5170	.0180	2.47	.1861	.1538
30.	21.63	2726.	2.7265	1.1663	.0407	2.34	.4017	.3470
40.	36.86	4646.	4.6462	2.1291	.0743	2.18	.6845	.6335
50.	53.85	6788.	6.7878	3.3607	.1173	2.02	1.0000	1.0000
60.	70.83	8928.	8.9282	4.8079	.1679	1.86	1.3153	1.4306
70.	86.90	10954.	10.9538	6.4231	.2242	1.71	1.6137	1.9112

α 1500.00		RPM						
LEAD	IHP	T	DMT	DMV	WS	DMT/DMV	NDMT	NDMV
20.	24.91	1047.	1.0466	.4281	.0448	2.45	.1786	.1496
30.	54.37	2284.	2.2845	.9765	.1023	2.34	.3899	.3413
40.	94.21	3958.	3.9584	1.8235	.1910	2.17	.6755	.6373
45.	116.25	4884.	4.8845	2.2988	.2408	2.12	.8336	.8035
50.	139.46	5860.	5.8597	2.8611	.2997	2.05	1.0000	1.0000
60.	185.09	7777.	7.7769	4.0969	.4291	1.90	1.3272	1.4319

α 2500.00		RPM						
LEAD	IHP	T	DMT	DMV	WS	DMT/DMV	NDMT	NDMV
20.	35.46	894.	.8940	.3655	.0638	2.45	.1733	.1468
30.	78.51	1979.	1.9792	.8421	.1470	2.35	.3837	.3383
35.	105.95	2671.	2.6710	1.1681	.2039	2.29	.5178	.4693
40.	137.16	3458.	3.4578	1.5596	.2722	2.22	.6704	.6265
45.	169.78	4280.	4.2802	1.9948	.3482	2.15	.8298	.8013
50.	204.61	5158.	5.1582	2.4893	.4346	2.07	1.0000	1.0000

Figure 2-5 - Expander D_{MT} and D_{MV} Calculation Results

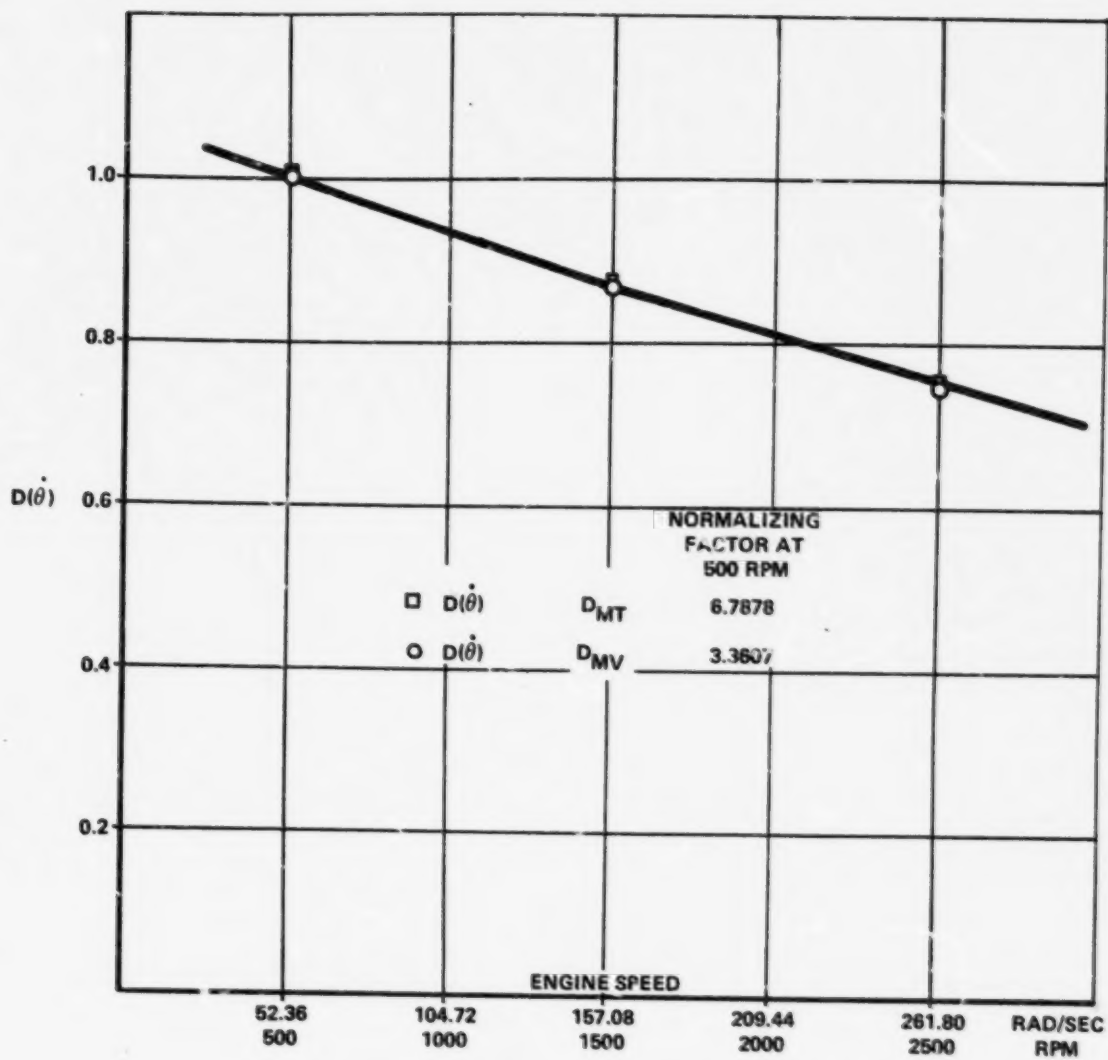


Figure 2-6 - Engine Speed Correction Curves

GRADE	00	PERCENT	13.16				
MPH FRICTION	RPM KJAD	RAD/SEC FAN	WST FEED P	BURNER	TOTAL	TORQUE	
5 .20	96.51 .96	10.106 1.55	.00706 .05	.66	3.45	2251.416	
10 .44	193.01 2.01	20.212 1.10	.00882 .06	.66	4.29	1401.833	
15 .71	289.52 3.20	30.318 .73	.01109 .08	.66	5.40	1175.712	
20 1.01	386.03 4.60	40.425 .44	.01404 .11	.66	6.63	1115.775	
25 1.35	482.53 6.26	50.531 .23	.01776 .14	.66	8.66	1130.766	
30 1.72	579.04 8.25	60.637 .10	.02245 .16	.66	10.93	1189.606	
35 2.12	675.55 10.64	70.743 .06	.02616 .23	.66	13.72	1279.630	
40 2.55	772.05 13.47	80.849 .09	.03507 .27	.66	17.09	1394.710	
45 3.02	868.56 16.83	90.955 .21	.04333 .36	.66	21.09	1530.626	
50 3.52	965.07 20.76	101.061 .41	.05301 .44	.66	25.61	1685.416	
55 4.05	1061.57 25.33	111.166 .66	.06427 .54	.69	31.29	1857.706	
60 4.61	1158.08 30.61	121.274 1.04	.07724 .65	.69	37.61	2046.605	
65 5.21	1254.59 36.65	131.380 1.49	.09205 .76	.70	44.62	2251.504	
70 5.84	1351.09 43.51	141.486 2.01	.10684 .92	.71	52.99	2471.796	
75 6.51	1447.60 51.27	151.592 2.61	.12774 1.06	.73	62.19	2707.618	
80 7.20	1544.11 59.97	161.698 3.30	.14869 1.26	.76	72.49	2953.602	
85 7.93	1640.61 69.69	171.804 4.06	.17242 1.46	.80	83.95	3224.572	
90 8.69	1737.12 80.48	181.911 4.91	.19848 1.66	.87	96.63	3506.030	
95 9.49	1833.63 92.41	192.017 5.64	.22721 1.93	.96	110.62	3802.349	
100 10.32	1930.13 105.54	202.123 6.64	.25877 2.20	1.09	125.99	4113.974	
WST	907.2911	122.703					

Figure 2-7 - Expander Load Tabulation - High Gear

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GRADE	00	PERCENT	20.5				
MPH	RPM	RAD/SEC	WST	BURNER	TOTAL	TORQUE	
FRICTION	ROAD	FAN	FEED P				
5	150.33	15.743	.00600				
.33	.96	1.29	.05	.66	3.31	1365.059	
10	300.67	31.466	.00660				
.74	2.01	.69	.06	.66	4.19	677.375	
15	451.00	47.229	.01127				
1.23	3.20	.29	.06	.66	5.49	766.924	
20	601.33	62.971	.01496				
1.60	4.60	.09	.12	.66	7.26	763.155	
25	751.67	78.714	.01976				
2.46	6.26	.08	.16	.66	9.63	607.361	
30	902.00	94.457	.02587				
3.19	8.25	.27	.21	.66	12.59	679.774	
35	1052.33	110.200	.03336				
4.00	10.64	.65	.27	.66	14.24	972.610	
40	1202.67	125.943	.04237				
4.89	13.47	1.24	.35	.66	20.63	1031.035	
45	1353.00	141.686	.05305				
5.65	16.83	2.02	.44	.66	25.63	1203.073	
50	1503.33	157.429	.06551				
6.90	20.76	3.00	.55	.69	31.90	1337.207	
55	1653.67	173.171	.07990				
6.03	25.33	4.17	.67	.69	36.90	1462.607	
60	1804.00	188.914	.09634				
9.24	30.61	5.54	.61	.70	46.91	1636.715	
65	1954.33	204.657	.11497				
10.53	36.65	7.11	.97	.72	55.98	1805.165	
70	2104.67	220.400	.13592				
11.90	43.51	8.66	1.15	.74	66.16	1961.721	
75	2255.00	236.143	.15934				
13.35	51.27	10.64	1.35	.78	77.56	2166.246	
80	2405.33	251.886	.18536				
14.87	59.97	13.00	1.57	.83	90.25	2364.683	
85	2555.67	267.629	.21413				
16.48	69.69	15.35	1.62	.91	104.25	2571.033	
90	2706.00	283.371	.24561				
18.17	80.48	17.91	2.09	1.03	119.66	2787.367	
95	2856.33	299.114	.28054				
19.93	92.41	20.66	2.36	1.20	136.59	3013.616	
100	3006.67	314.657	.31651				
21.78	105.54	23.60	2.70	1.44	155.07	3250.576	
WST	1115.9786	150.9261					
32.6							

Figure 2-8 - Expander Load Tabulation - Second Gear

NEXT? 3
 ORDER? 2
 CJEFF

VALUE

0	1753.54
1	-15.0552
2	.136385

NEXT? 5
 ORDER? 2

JBS X	JBS Y	CALC Y	DIFF
10.1061	2251.42	1615.32	636.102
20.2123	1401.63	1504.95	-103.12
30.3184	1175.71	1422.45	-246.74
40.4246	1115.78	1367.81	-252.034
50.5307	1130.57	1341.03	-210.457
60.6369	1189.61	1342.1	-152.494
70.743	1279.83	1371.04	-91.2053
80.8492	1394.71	1427.63	-33.1169
90.9553	1530.63	1512.46	18.1453
101.062	1685.42	1625.	60.4204
111.168	1857.71	1765.37	92.3412
121.274	2046.6	1933.6	113.007
131.38	2251.5	2129.69	121.817
141.486	2472.	2353.64	118.362
151.592	2707.82	2605.45	102.373
161.698	2958.8	2885.11	73.6899
171.805	3224.87	3192.64	32.232
181.911	3506.03	3526.03	-21.9947
192.017	3802.35	3891.27	-86.9236
202.123	4113.97	4282.38	-166.401

NEXT? 7

ENTER ORDER, STARTING VALUE, ENDING VALUE, INTERVAL? 2, 0, 300, 20
 X Y

0	1753.54
20	1506.99
40	1369.54
60	1341.21
80	1421.99
100	1611.67
120	1910.86
140	2316.96
160	2836.17
180	3462.48
200	4197.91
220	5042.44
240	5996.08
260	7058.83
280	8230.69
300	9511.66

Figure 2-9 - Second Order Load Curve Fitting - High Gear

COEFF	VALUE
0	1034.99
1	-4.03255
2	3.62334 E-2

NEXT? 5
JORDER? 2

JBS X	JBS Y	CALC Y	DIFF
15.7429	1386.06	930.484	407.575
31.4657	677.375	943.941	-66.5658
47.2266	766.924	925.357	-158.433
62.9714	763.155	924.733	-161.576
78.7143	807.361	942.069	-134.708
94.4571	879.974	977.365	-97.3915
110.2	972.61	1030.62	-58.0111
125.943	1061.08	1101.84	-20.753
141.666	1203.08	1191.01	12.064
157.429	1337.21	1298.15	39.0572
173.171	1462.61	1423.25	59.3603
188.914	1636.71	1566.3	72.4124
204.657	1805.16	1727.32	77.3466
220.4	1961.72	1906.29	75.4269
236.143	2165.25	2103.23	65.0176
251.886	2364.66	2318.13	46.5562
267.629	2571.03	2550.98	20.0495
283.371	2767.37	2801.6	-14.4313
299.114	3013.62	3070.55	-56.7596
314.857	3250.58	3357.31	-106.735

NEXT? 7
ENTER JORDER, STARTING VALUE, ENDING VALUE, INTERVAL? 2, 0, 300, 20

X	Y
0	1034.99
20	963.331
40	931.66
60	923.476
80	944.276
100	994.067
120	1072.84
140	1160.61
160	1317.36
180	1483.09
200	1677.82
220	1901.52
240	2154.22
260	2435.9
280	2746.57
300	3086.23



Figure 2-10 - Second Order Load Curve Fitting - Second Gear

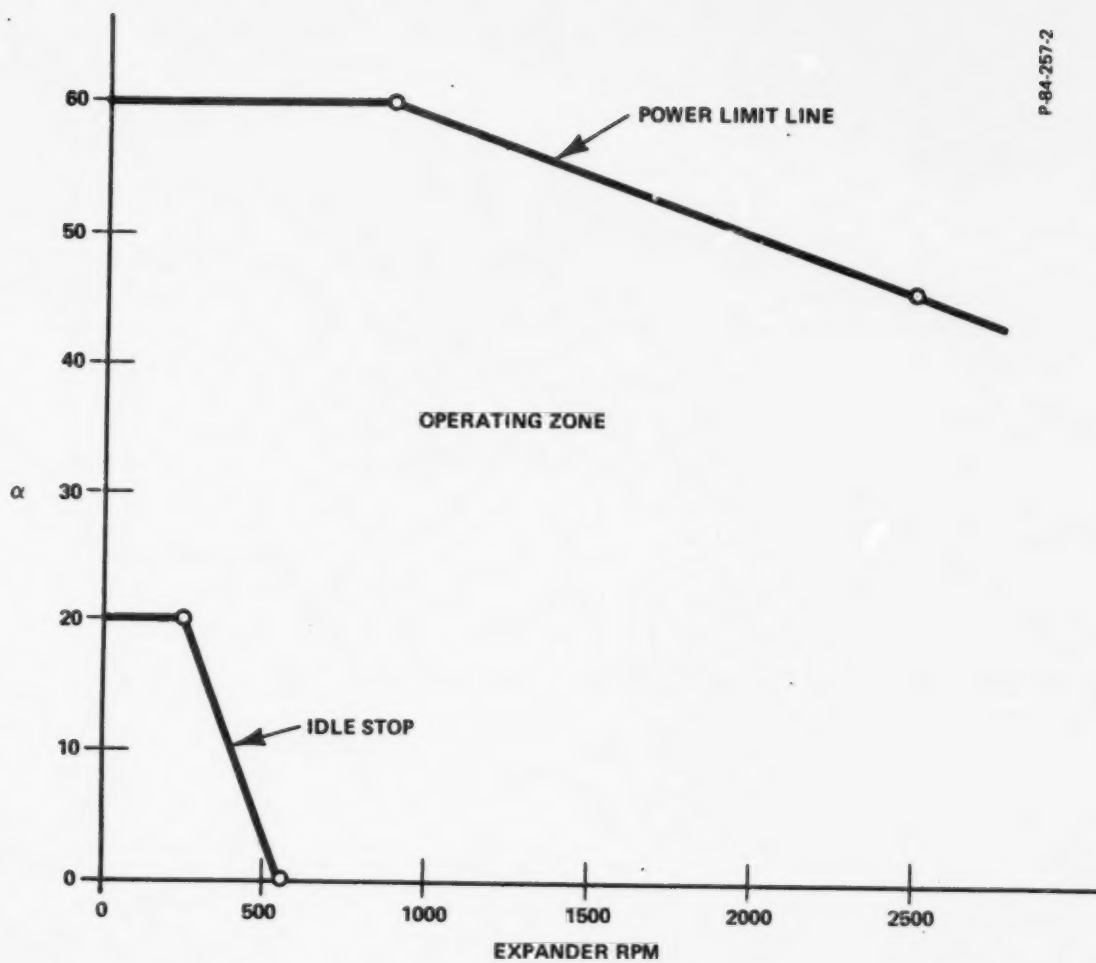


Figure 2-11 - Power Limiter and Idle Stop

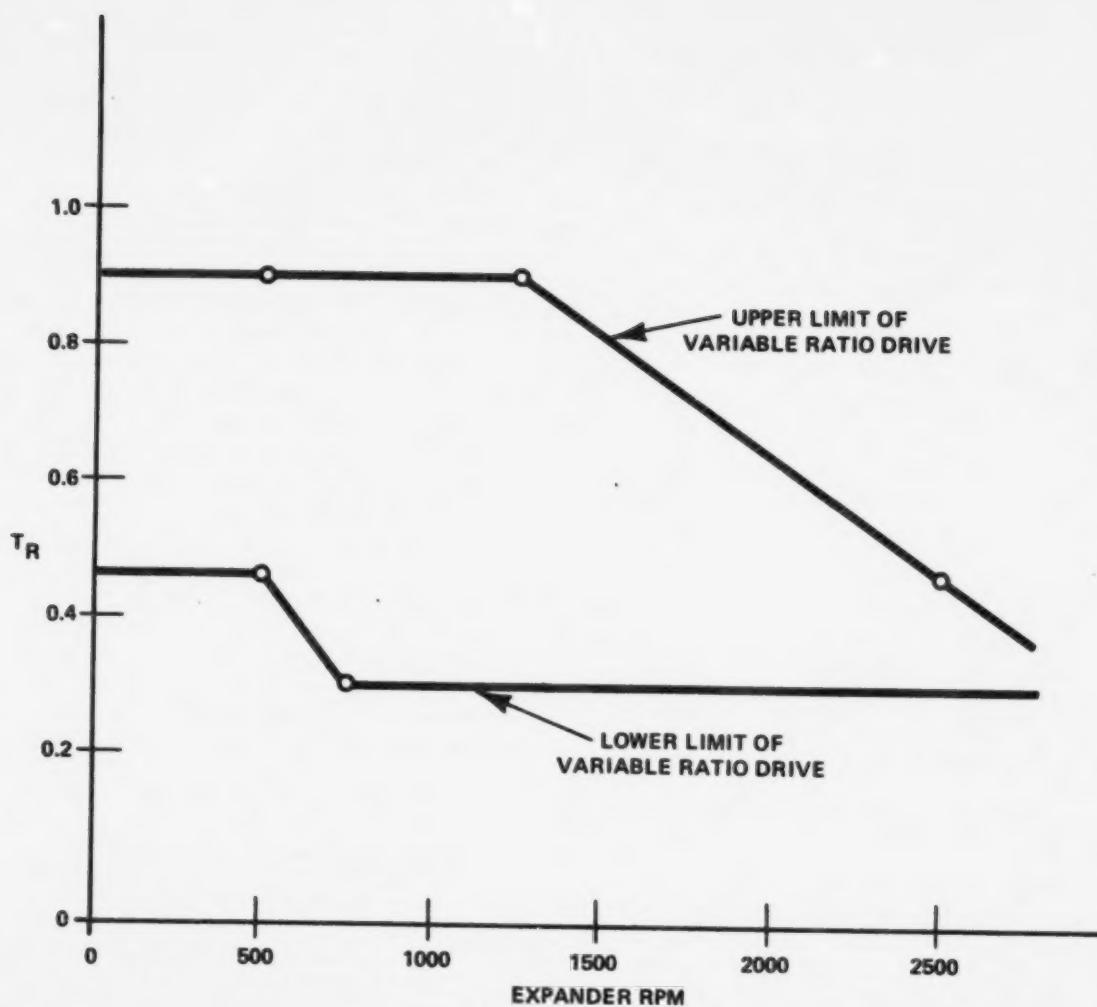


Figure 2-12 - Variable Drive Ratio Limits

SECTION 3

HYBRID COMPUTER RESULTS

The details of the hybrid computer model are given in Section 2. During January, information describing changes in the SES design of the Mark 5 steam generator and also changes in the transmission gearing were received by Bendix. The major effects of the new constants are demonstrated in the changes of the reflected inertia at the engine and are due to the gearing changes. The insertion of an additional pass in the steam generator modified the heat transfer coefficients used in the simulation. In addition the added metal affected the time constant of the steam generator to a degree. Other effects included a faster responding bare-tube superheater pass and changes of the gas flow path to the counterflow system shown in Figure 3-1. A number of control concepts were evaluated prior to the implementation of these changes. As the results have general validity, a short summary of these results obtained during February 1972 are discussed in the following.

The hybrid computer model of the steam generator was integrated with the vehicle and engine driven auxiliaries. The general schematic of the integrated system is shown in Figure 3-1. The information flow diagram includes a very simple predictive controller that sets the variable speed auxiliary drive ratio on the basis of throttle position and engine speed. The pressure trim controller is a simple direct-acting bypass valve. The temperature trim control is a position servo actuated damper in series with the combustion air blower. The model was used to establish preliminary gain settings that provide stable operation in the pressure and temperature trim control loops and also to evaluate the operating characteristics of several bypass feedwater control designs.

3.1 HYBRID RESULTS OF CROSS-COUNTER FLOW STEAM GENERATOR CONFIGURATION

Effect of Prediction Accuracy

Initial evaluations were conducted using an idealized predictive flow control system. In this idealized case it was assumed that rapid response (2 to 5 hertz bandpass) fuel and feedwater controller would be able to match the demanded fuel flow and feedwater flow commanded by the predictive controller. Using this idealized model of the predictive control system the required prediction accuracy was evaluated. Results indicated that an 80 percent accuracy in the prediction is sufficient, and trim control operation was able to handle as much as 50 percent error in the predicted signal. With zero prediction, in other words with straight closed-loop control, pressure transients were controlled sufficiently by a bypass controller. However, the temperature errors in the non-predictive case became excessively large. This initial evaluation

of the control concept also indicated the need of combustion air and feedwater pump capacity in excess of the steady state requirements. This excess is mainly needed during acceleration. In the feed pump circuit a sudden increase in the accelerator pedal may open the inlet valve to the expander at a rate faster than the feedwater system can follow. If no excess pumping capacity is available, then the initial steam pressure droop exists for an extended period of time. This reduces the maximum torque capability of the expander and limits the acceleration of the car.

On-Off Pressure Control

Operation of an on-off pressure controller was also evaluated. This pressure concept offers a very simple mechanization of the bypass concept. Over the required flow range of 20 to 1 excessive limit cycling was present. Pressure deadband was varied from 990 to 1010 psi for a narrow range design to 900 to 1100 psi for a wide range deadband evaluation. The speed of the bypass valve was slow and was varied from two to four seconds end-to-end travel time. Under these operating conditions excessive pressure variations were present. The relative slow response bypass valve was to simulate an electrically operated bypass valve. The extreme limit cycle was evident at all conditions evaluated.

Proportional Pressure Control

An optimization of the bypass feedwater control when combined with the simple predictive control of engine driven auxiliaries was carried out. This combination may offer the simplest and least expensive control configuration of the feedwater control system. Typical results describing this optimization study are given in Figures 3-2 through 3-7. The examples are given in order to illustrate the use of the hybrid computer model for the control system optimization. In this type of control system interaction is present between feedwater and fuel control. In addition, the relative capacity of combustion air blower and feedpump as compared with steady state engine demand also influences the design and the performance of the power plant controls. A wide range of choices are practical in these areas. At the present time the choices were not optimized. However, results presented are typical of the type of controls proposed for the prototype control mode.

Slow Bypass Valve

Figures 3-2 and 3-3 describe the operation of the bypass pressure control using a relatively slow bypass valve, requiring two seconds to travel from the fully opened to fully closed position. The two figures represent simultaneous recordings of 16 selected variables in the hybrid computer simulation. The designations of the variables can be correlated with the schematic (Figure 3-1). Pressure control characteristics of the feedwater system using a slow response bypass valve are described in Figure 3-2 and 3-3. The traces describe the response of the steam generator and the control system to a step change in throttle setting

corresponding to approximately 40-70 mph vehicle speed range with the corresponding steam flow rates from 0.315 to 0.415 pounds per second. Wide excursion in steam pressure occurs during the transient. The slowly responding bypass valve is not able to cope with the rapidly varying steam flow. Pressure excursions of plus and minus 500 psi occur in this case. A slow temperature transient follows the pressure transient. Steady state steam pressure varies from 1200 to 1020 psi. The offset is due to the fact that pump capacity is significantly larger than steady state engine requirements. The bypass valve opening changes from 40 percent to 1 percent opening. In general, this indicates that at a minimum engine speed, low excess feedwater and combustion air flow is available. Figure 3-3 shows the corresponding metal temperature transients and boiler pressure transients. In addition, it may be seen that the predictive circuit changes the variable speed drive ratio from 0.64 to 1.20 during the transient.

Fast Linear Area Bypass Valve

A sufficient reduction of the pressure transient is achieved through the use of a faster bypass valve. Results are shown in Figures 3-4 and 3-5 for the new bypass valve design. The details of this design are shown in Section 5 under feedwater control. The traces indicate a linear scheduling between valve position and valve opening area. The results indicated that this type of a valve operates satisfactorily over a range of feedwater flow levels. However, at the very low flows, at the order of 3-5 percent of the full steam generator capacity, the gain of this type of a feedwater bypass valve becomes too high and unstable operation results. The steam pressure is controlled between 1000 and 960 psi during the transient. The maximum pressure excursion is 1010 psi and indicates a significant improvement over the previous design. The bypass opening varies from 50 to 10 percent opening. The variable drive transmission ratio varies from 1.37 to 1.55 during the transient shown. The steam flow steady state levels vary from 0.16 to 0.015 for this transient.

Fast Equal Percentage Bypass Valve

The performance of the feedwater control system can be improved at the low flow levels through the use of an equal percentage valve opening. In this configuration the area of the bypass valve varies approximately as the square of the valve position. This can be achieved by the use of a taper plug design as described again in Section 5. This is the configuration that was selected for experimental evaluation for the facsimile system. Typical results are again shown in Figures 3-6 and 3-7. Typical operation is shown for a steam flow change from 0.195 to 0.007 pounds per second. The minimum flow level is 1/50th of the rated 100 percent flow capacity of the SES-5 boiler. The steam pressure is controlled between 1020 to 960 psi. The variable transmission drive ratio is changed during the transient from 1.42 to 0.45. It appears that satisfactory steam

generator pressure control can be achieved through the use of the equal percentage bypass flow controller.

The above results are shown mainly to illustrate the use of the hybrid computer in the optimization of control. However, in the selected traces, the steam temperature is too high. This was due to the relatively low proportional gain used in the trim temperature control. The high excess combustion air blower capacity required a nearly 50 percent closing of the damper. It became apparent during the evaluations that capacity matches between the combustion air blower delivery and the feedwater pump delivery must be carefully observed. The speed-flow characteristic of the feedwater pump is essentially linear, while the blower output-speed curve is highly nonlinear in the operating range. This suggests that for the blower characteristics currently used it may be necessary to add a predictive signal to the air damper control in order to maintain an improved steam temperature control.

3.2 HYBRID RESULTS OF COUNTERFLOW STEAM GENERATOR CONFIGURATION

The hybrid computer simulation was modified during the reporting period to accommodate the changes due to the new SES-5 design of the steam generator. The effect of the additional fifth pass, which is a bare-tube superheater section, and the change of the gas flow path that locates the new section directly at the burner phase were evaluated. The evaluation of the control design for the new configuration is not completed at this time. Perhaps the most significant effect stems from the differences between blower and pump output characteristics as a function of auxiliary driveshaft speeds. In general, combustion air blowers of the type considered for the present application have a strong exponential characteristic of flow versus shaft speed. On the other hand, positive displacement pumps have a nearly linear output versus speed characteristics. When additional restraints are imposed on the control design, then the design of the temperature and pressure control for the steam generator becomes increasingly more complex. An example of such an additional restraint is the need for a minimum differential pressure drop across the combustion air circuit in order to reduce its sensitivity to ambient pressure disturbances at the inlet or exhaust of the combustor. A large number of parametric evaluations becomes essential to optimize the temperature and pressure control systems. This study is currently in progress and in the following, only a few examples of results will be given to illustrate some of the more interesting characteristics of the steam generator control system.

The need for limiting the maximum intake valve or cut-off angle as a function of engine RPM is illustrated in the traces shown in Figures 3-8 and 3-9. Again, the same sixteen selected variables are shown as a function of time for a step input in cut-off angle. When the steam demand of the expander becomes larger than the output capability of the steam generator, an additional demand for power results in a drop of both steam pressure and steam temperature. In the particular transient

shown, the steam pressure drops to 850 psi and the steam temperature drops to 650°F, from the nominal 1,000 psi and 1,000°F operating points. At the same time, in spite of the increase in intake valve settings, the expander speed remains at essentially the same level. The figures illustrate the definite need for a speed-sensitive limit on the maximum intake valve position. It is believed that for practical reasons this limit should be kept slightly under the maximum capacity of the steam generator output capacity. This choice will help to accommodate slight degradation of heat transfer coefficients during the life of the power plant.

The interaction between the temperature and pressure control system is illustrated in Figures 3-10 through 3-13. In both instances the steam generator is operated at a low level. At this point the system time constants both in the pressure and temperature control loop become the longest. Closed-loop control under these conditions becomes the most difficult due to the slow but strong interaction between the temperature and pressure control system. In both cases a step change in engine demand is initiated by changing the cut-off position alpha. In both cases, the feedwater flow input is manually controlled to maintain the steam pressure at 1,000 psi, with a lower gain in the temperature trim control. As shown in Figures 3-10 and 3-11, the control of the pressure becomes very difficult as the changes in feedwater flow affect both the outlet pressure and temperature. When the temperature control is improved as shown in Figures 3-12 and 3-13, pressure control becomes less affected by temperature changes, thus significantly improved control over the outlet pressure can be achieved even at very low operating levels. The increased level of air damper activity is a direct result of the increased temperature gain. It is apparent that good prediction accuracy will be required at the low power levels to reduce the air damper activity and the resultant modulation of the combustion gas flow at the minimum operating levels. The examples shown are characteristics of the general results of the hybrid computer simulation which appears to offer a very realistic tool for the optimization of the steam generator control system.

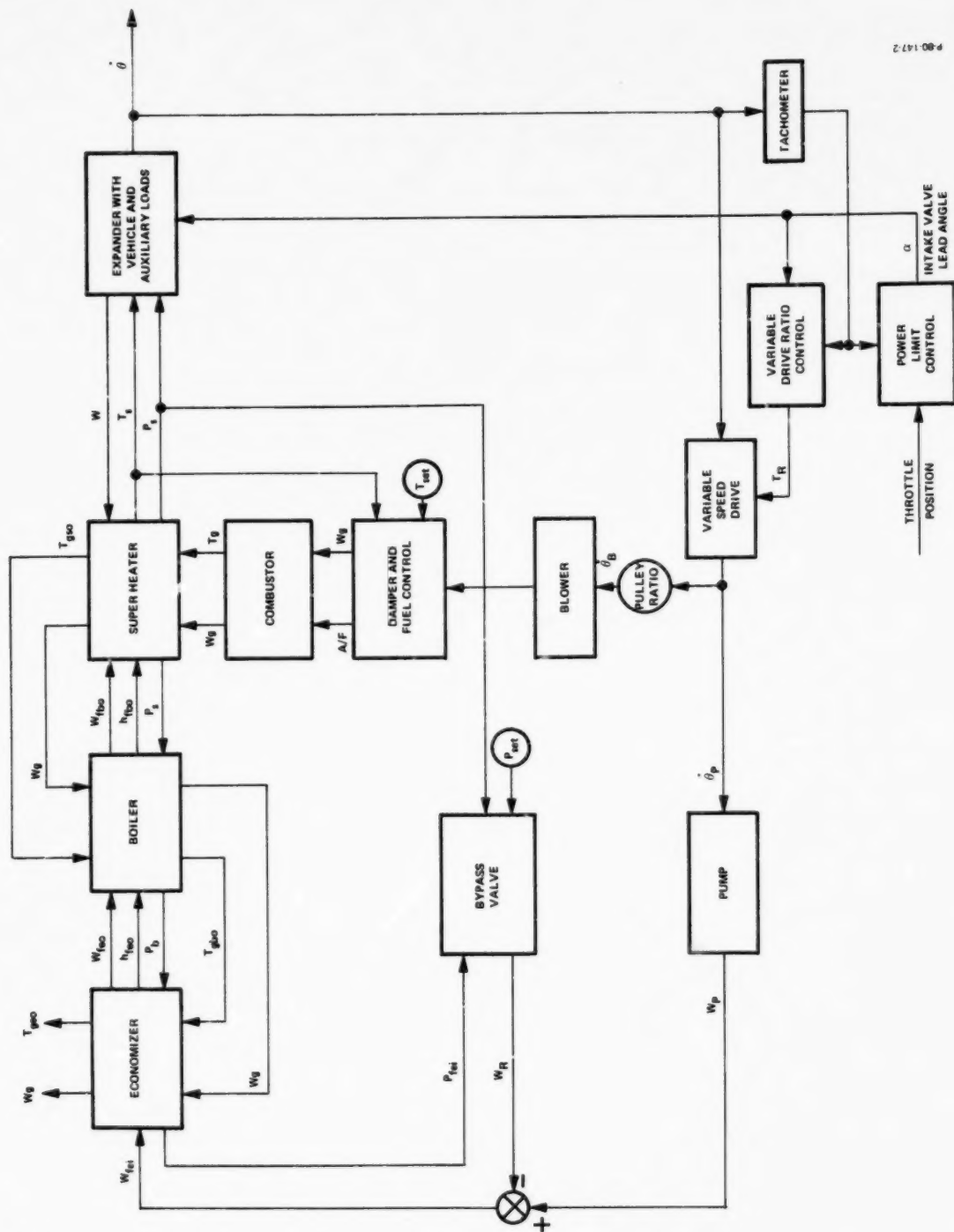


Figure 3-1 - Block Diagram of Hybrid Simulation;
Engine Driven Auxiliaries and Closed-Loop Control

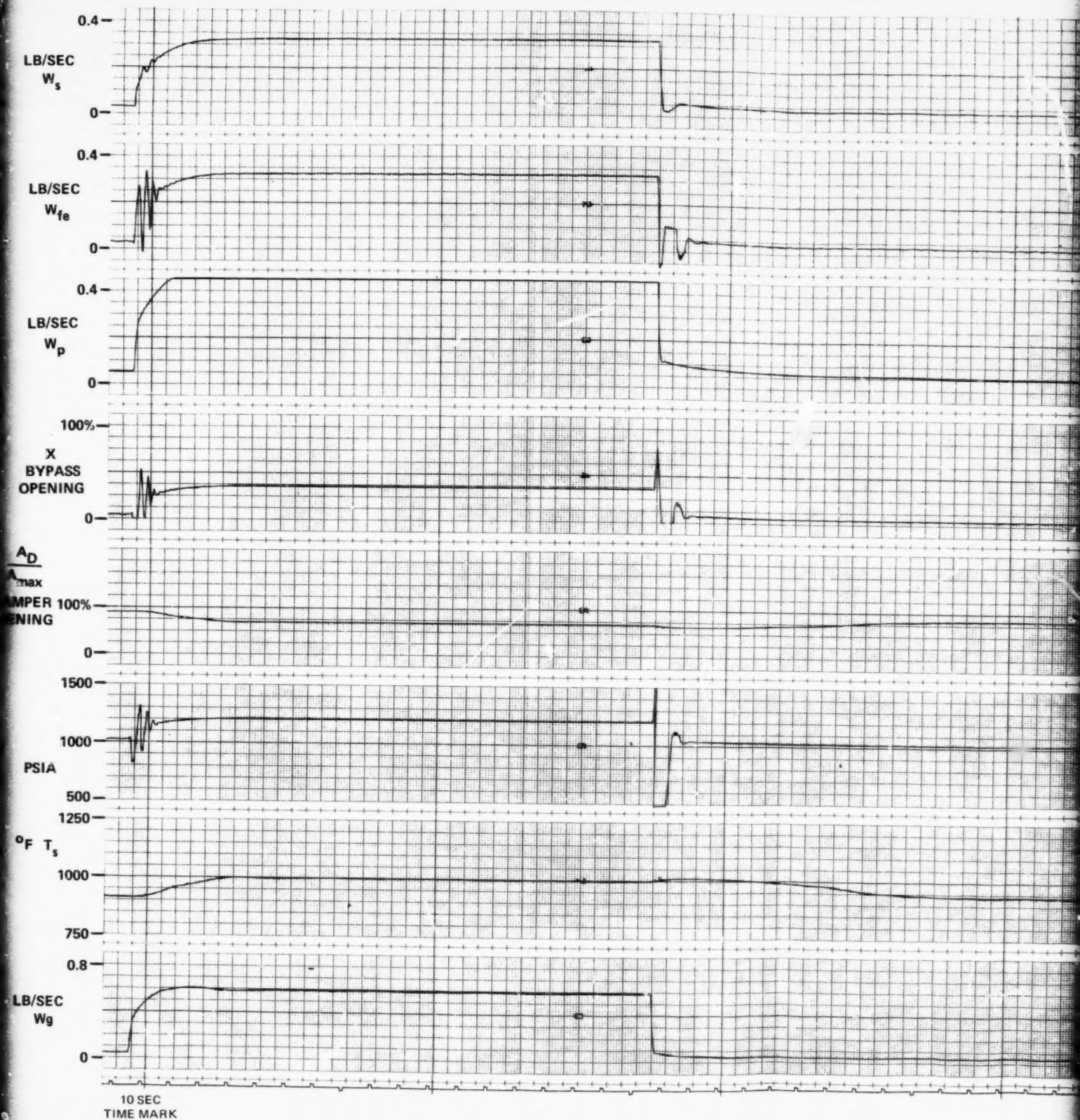


Figure 3-2 - Hybrid Computer Results; Slow Bypass Valve

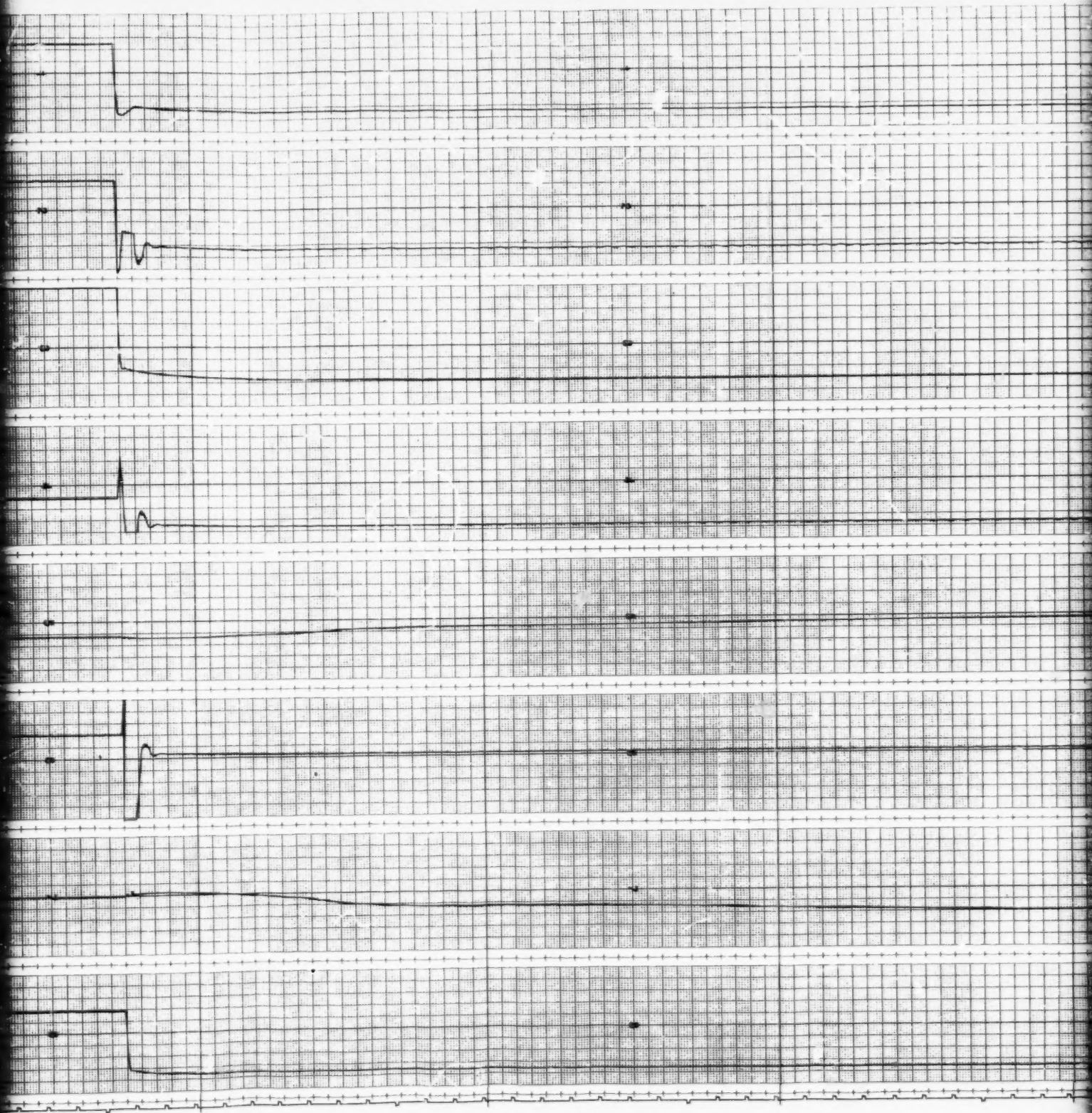
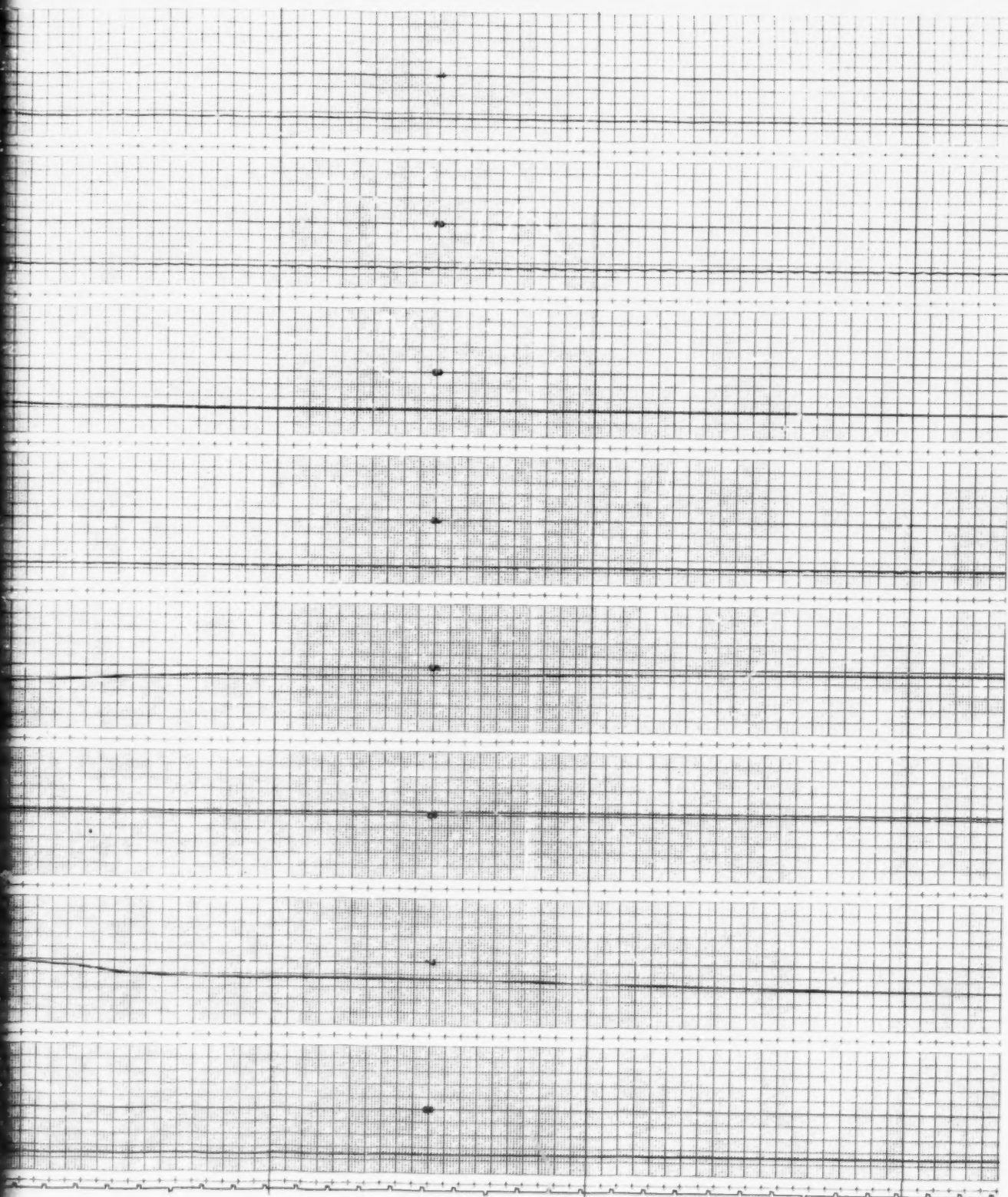


Figure 3-2 - Hybrid Computer Results; Slow Bypass Valve



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Computer Results; Slow Bypass Valve

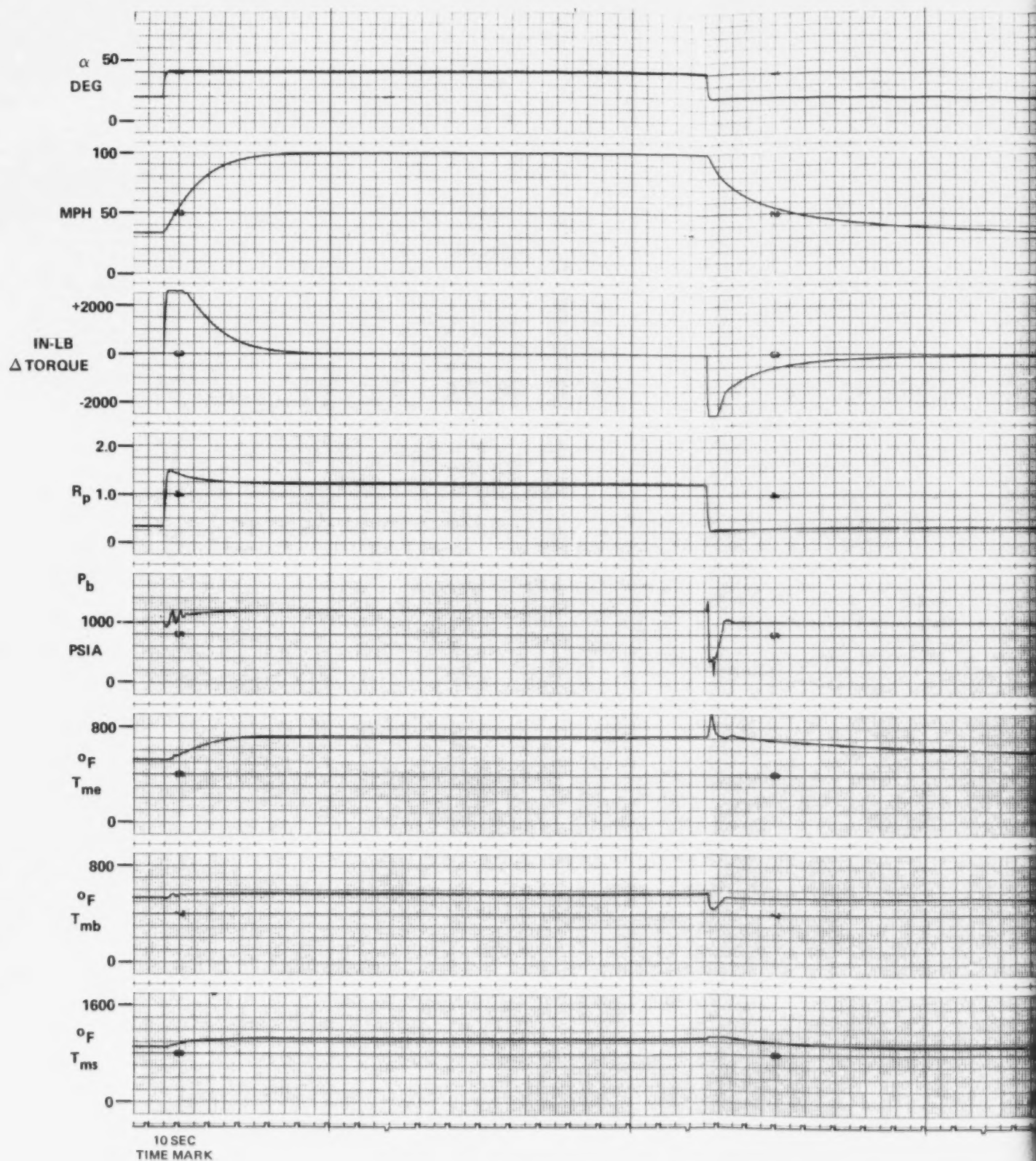
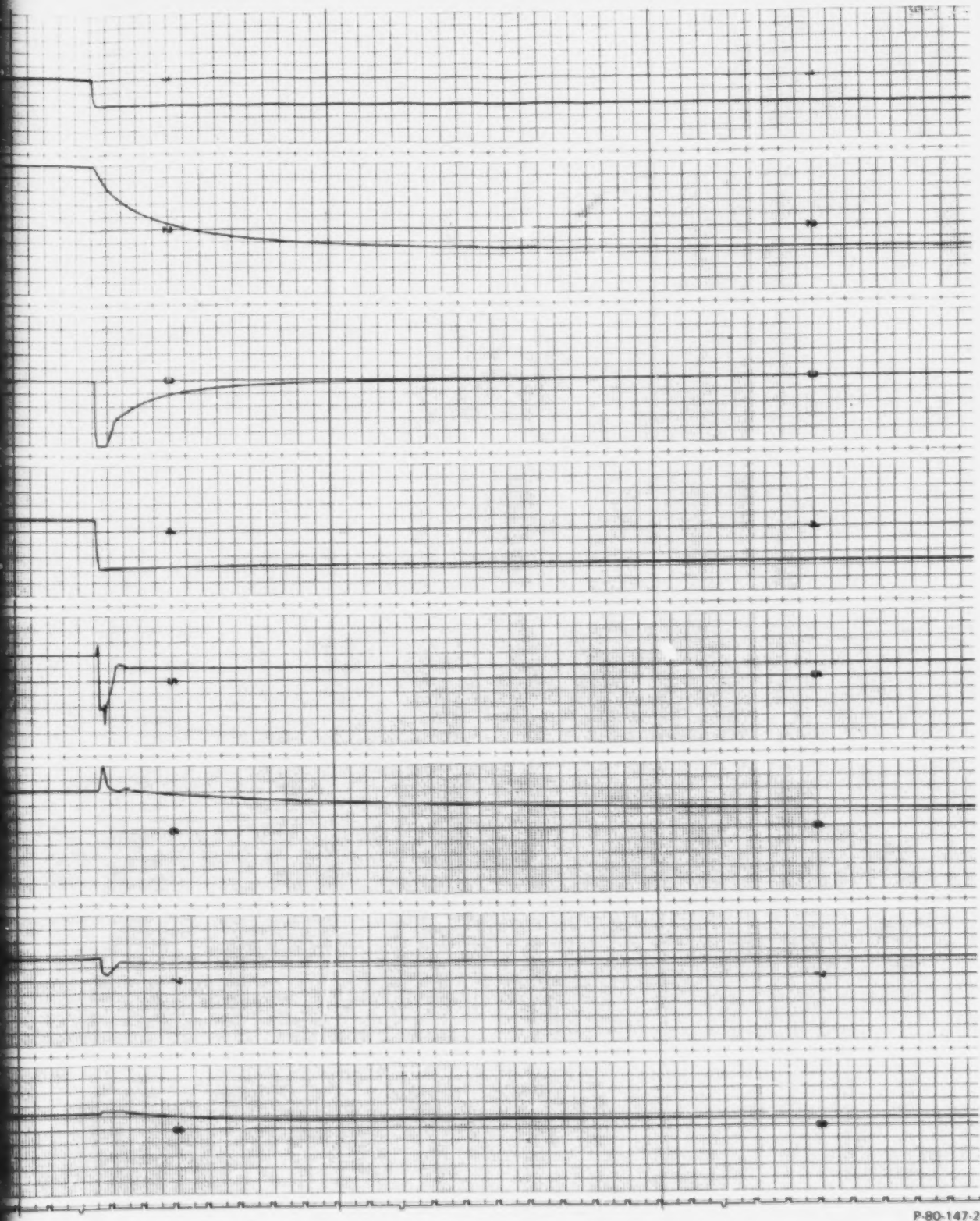


Figure 3-3 - Hybrid Computer Results; Slow Bypass Valve



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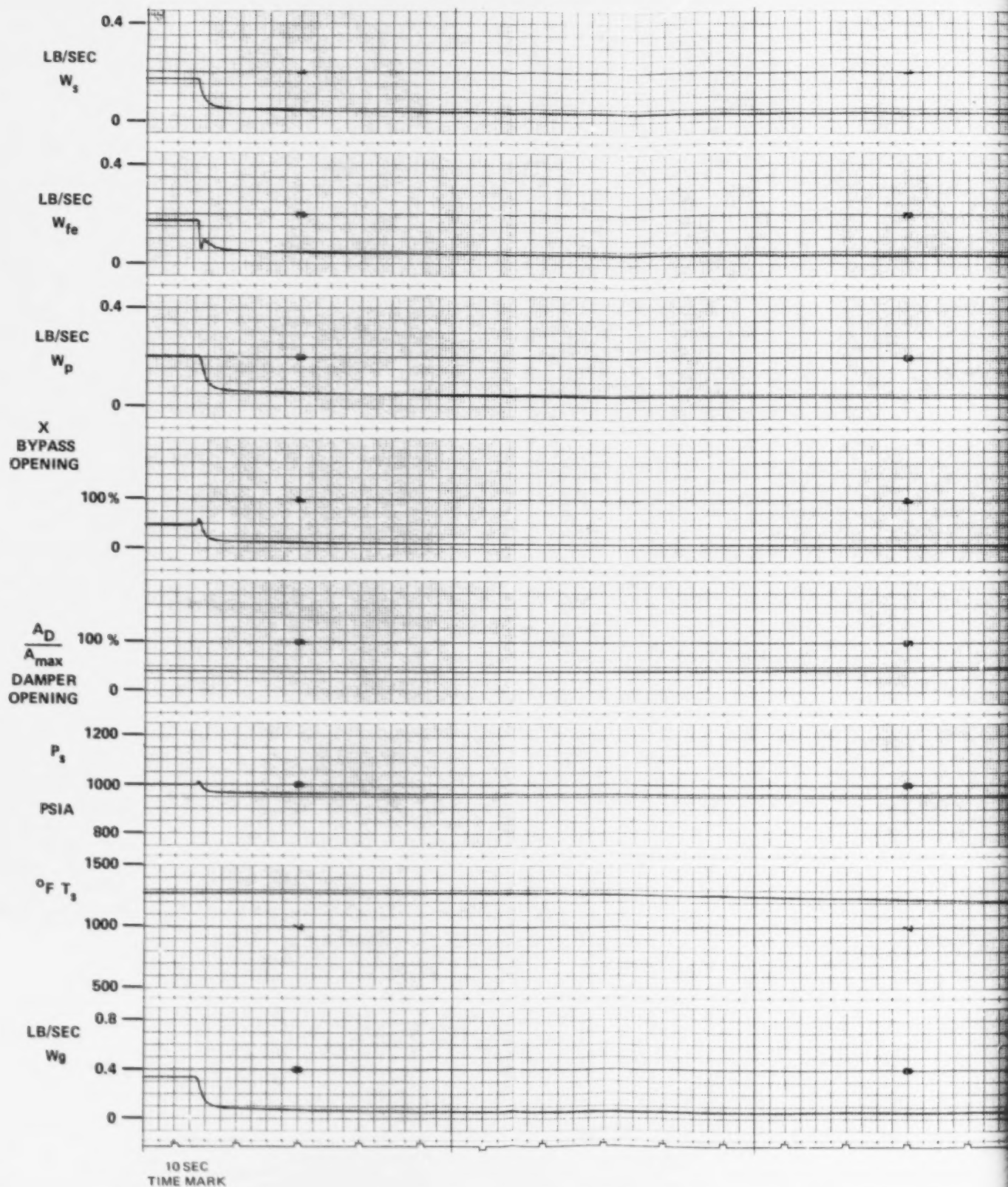
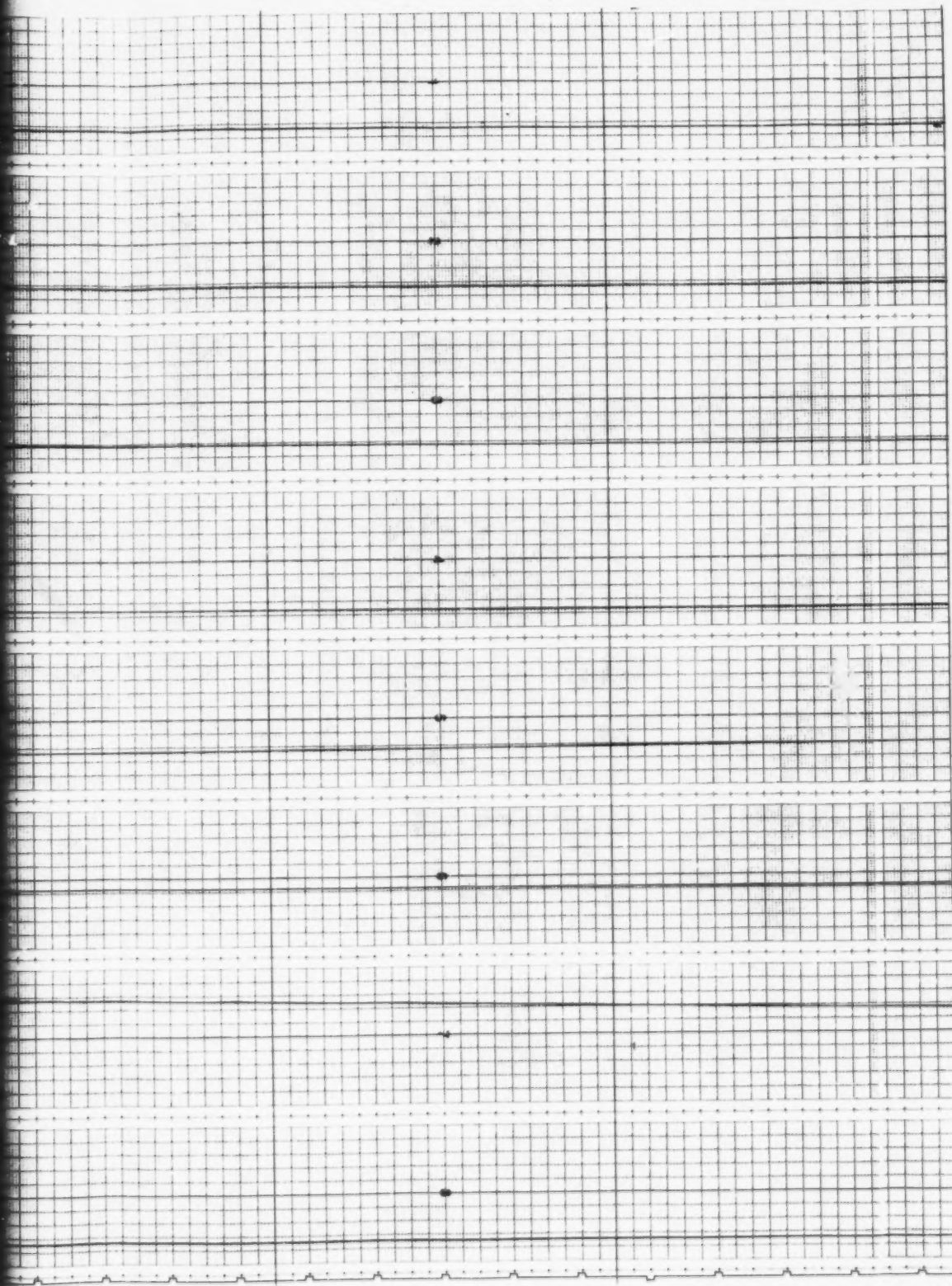


Figure 3-4 - Hybrid Computer Results; Fast Linear Bypass Valve



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α
DEG

$\dot{\theta}$
RAD/SEC

IN/LB
 Δ TORQUE

F

P_r
PS

σ_f
T_r

σ
T_r

σ
T_r

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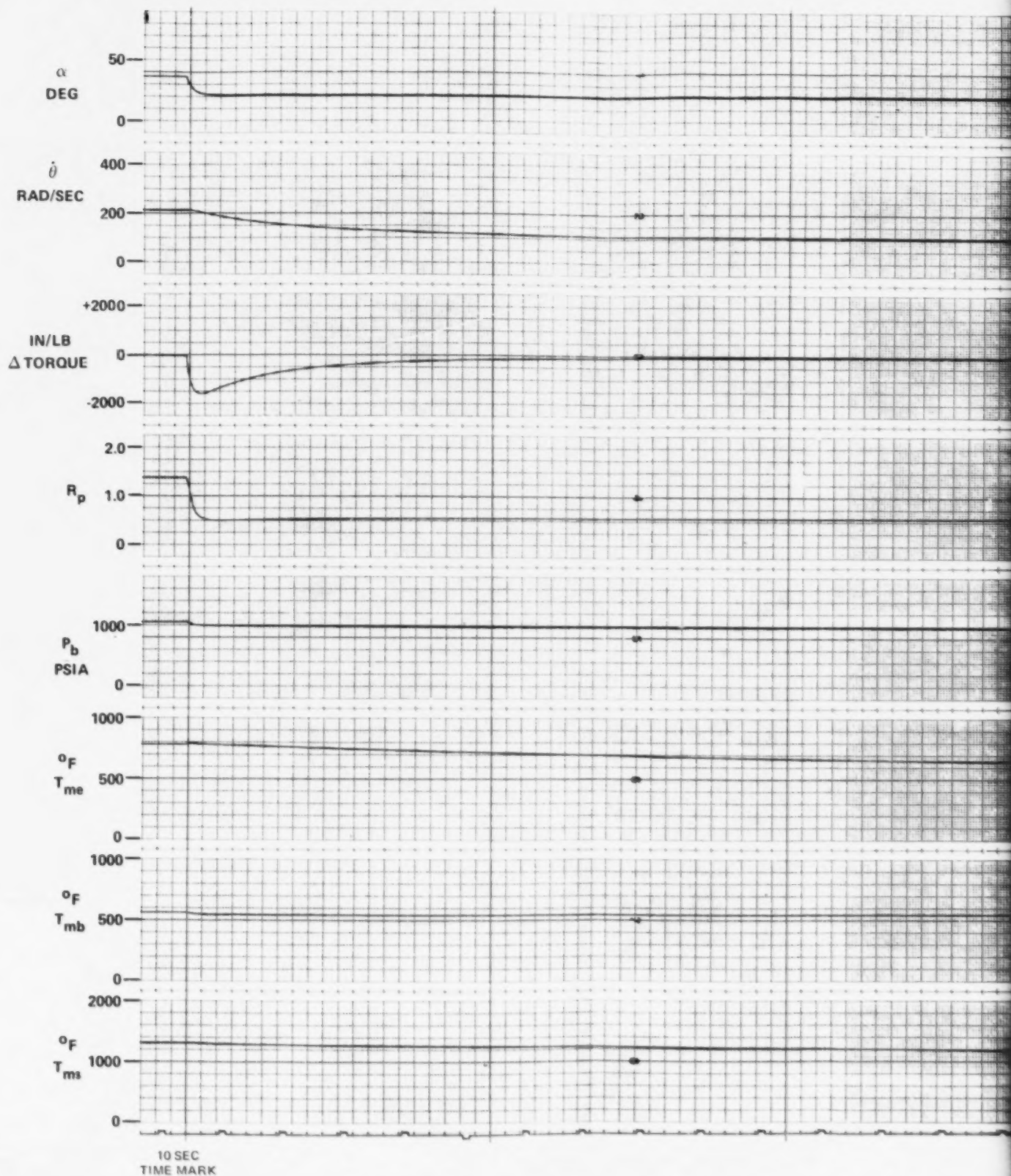
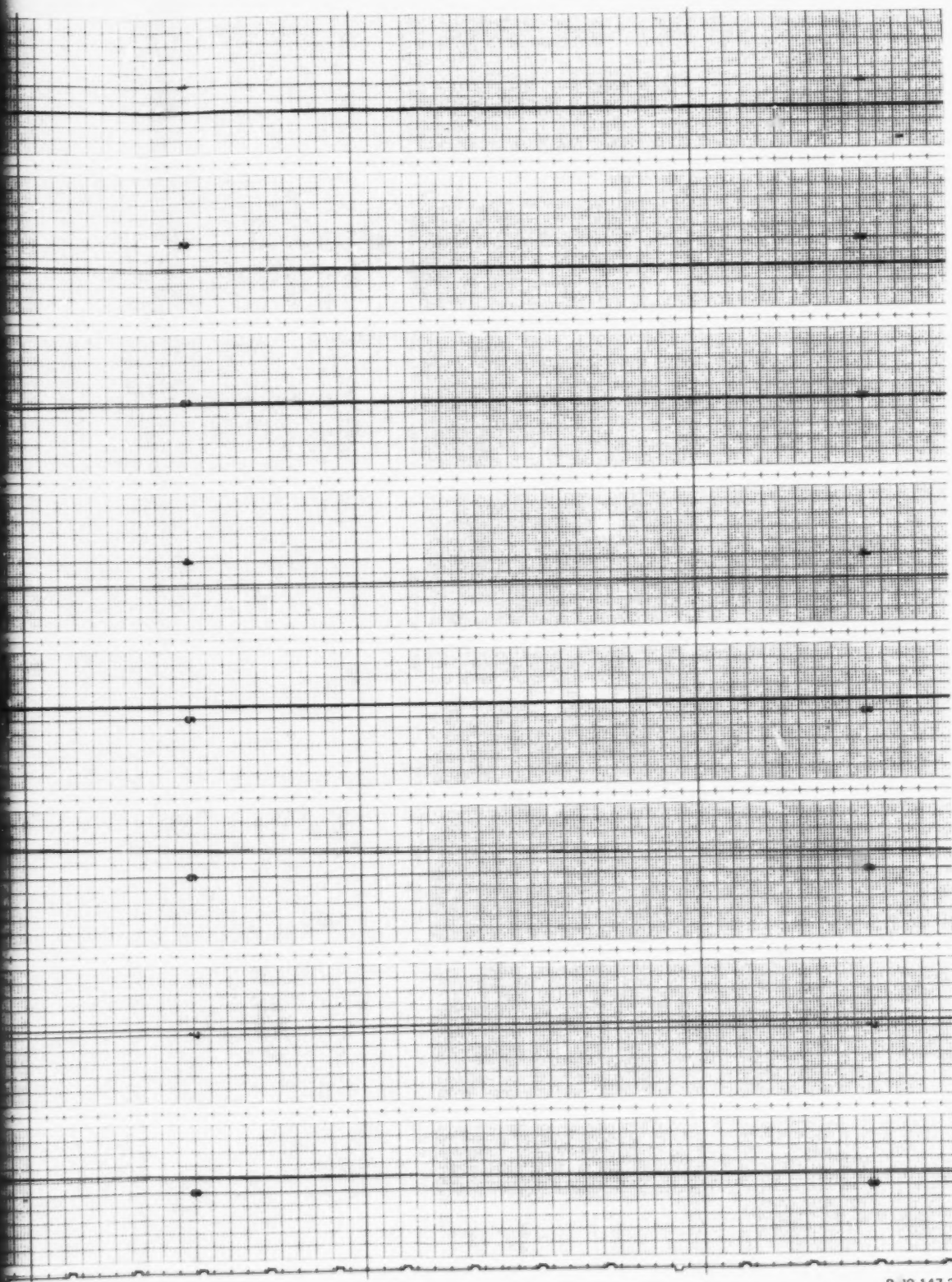


Figure 3-5 - Hybrid Computer Results; Fast Linear Bypass Valve



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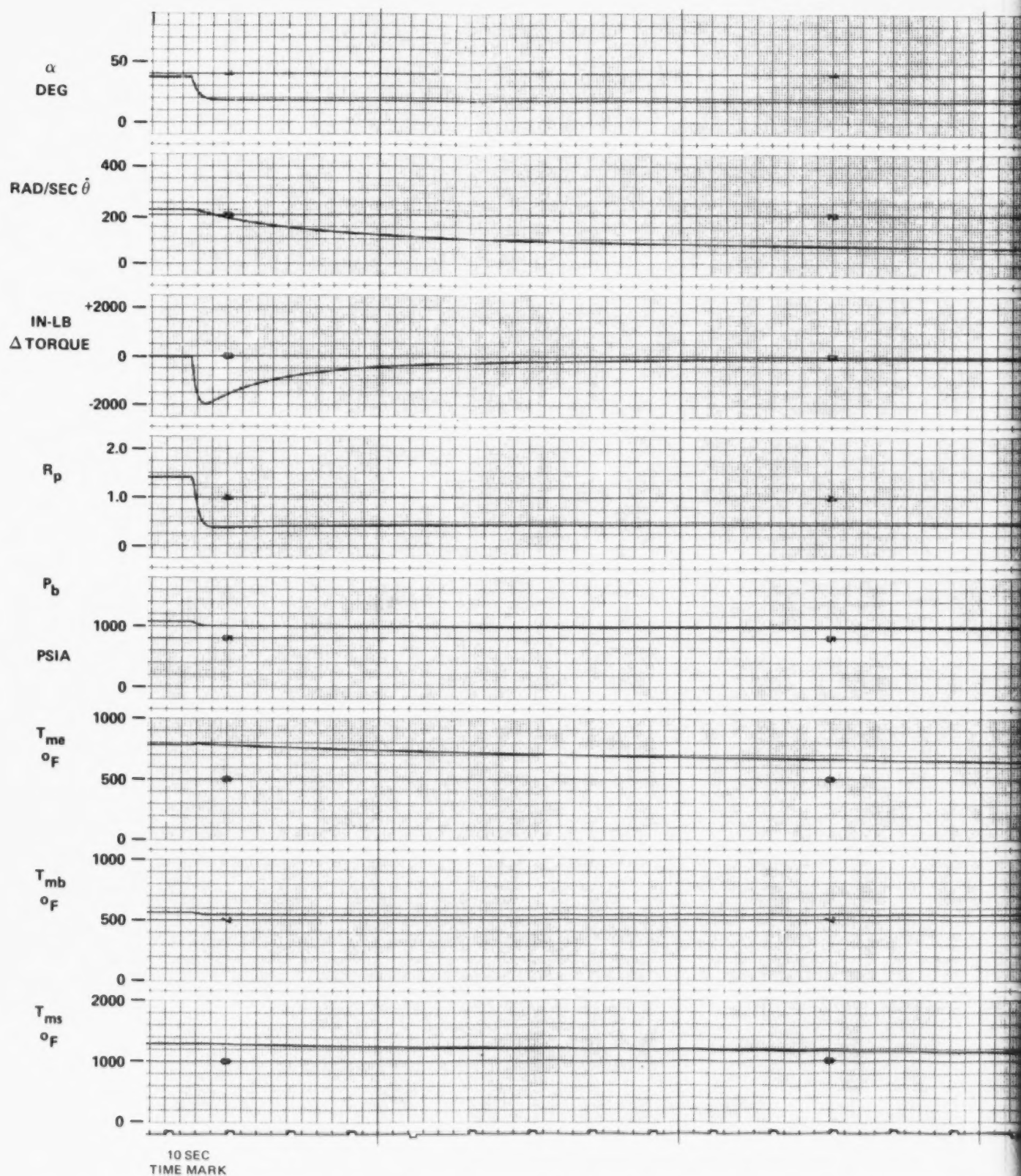
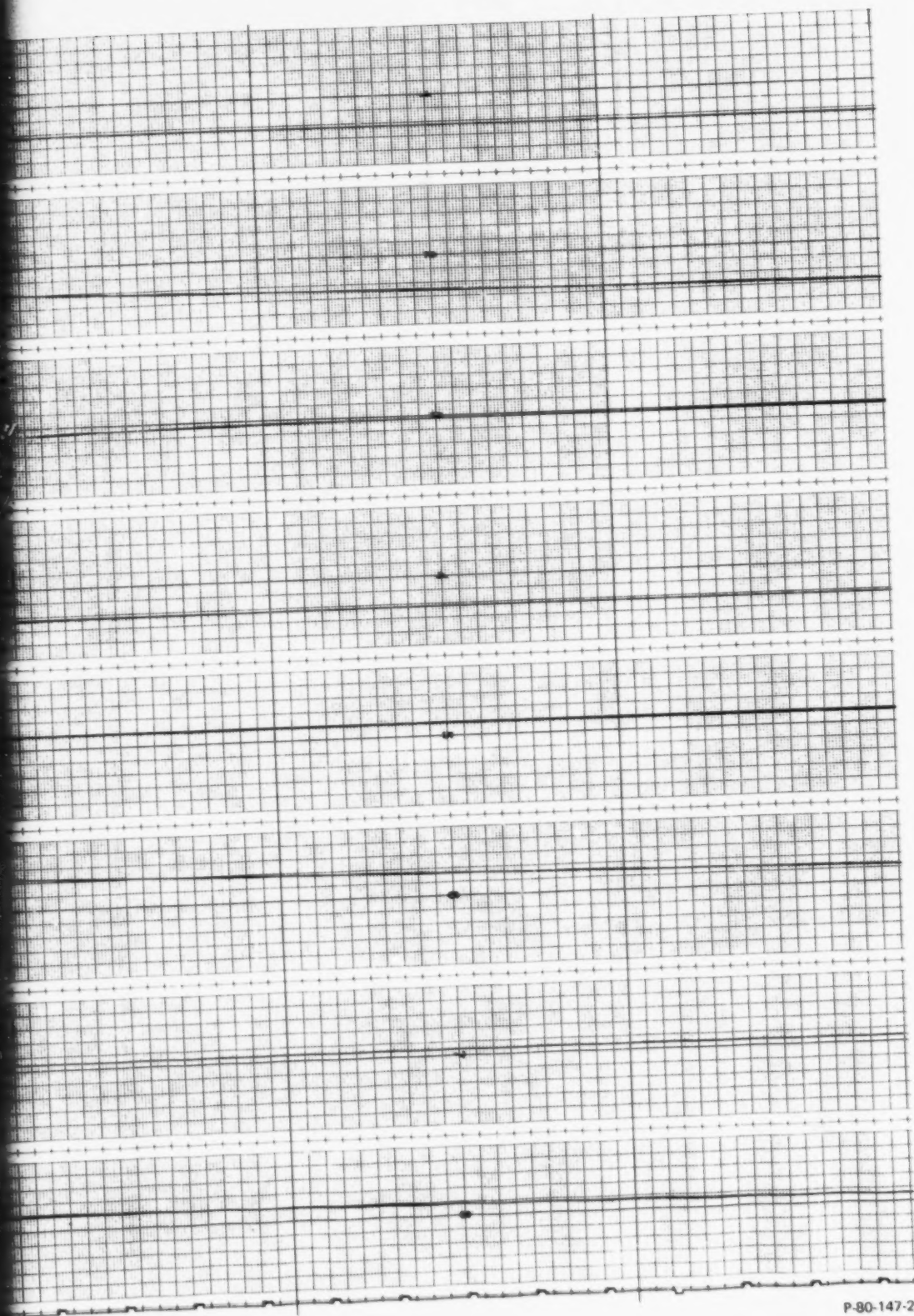


Figure 3-7 - Hybrid Computer Results; Fast Equal Percentage Bypass Valve



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Mid Computer Results; Fast Equal Percentage Bypass Valve

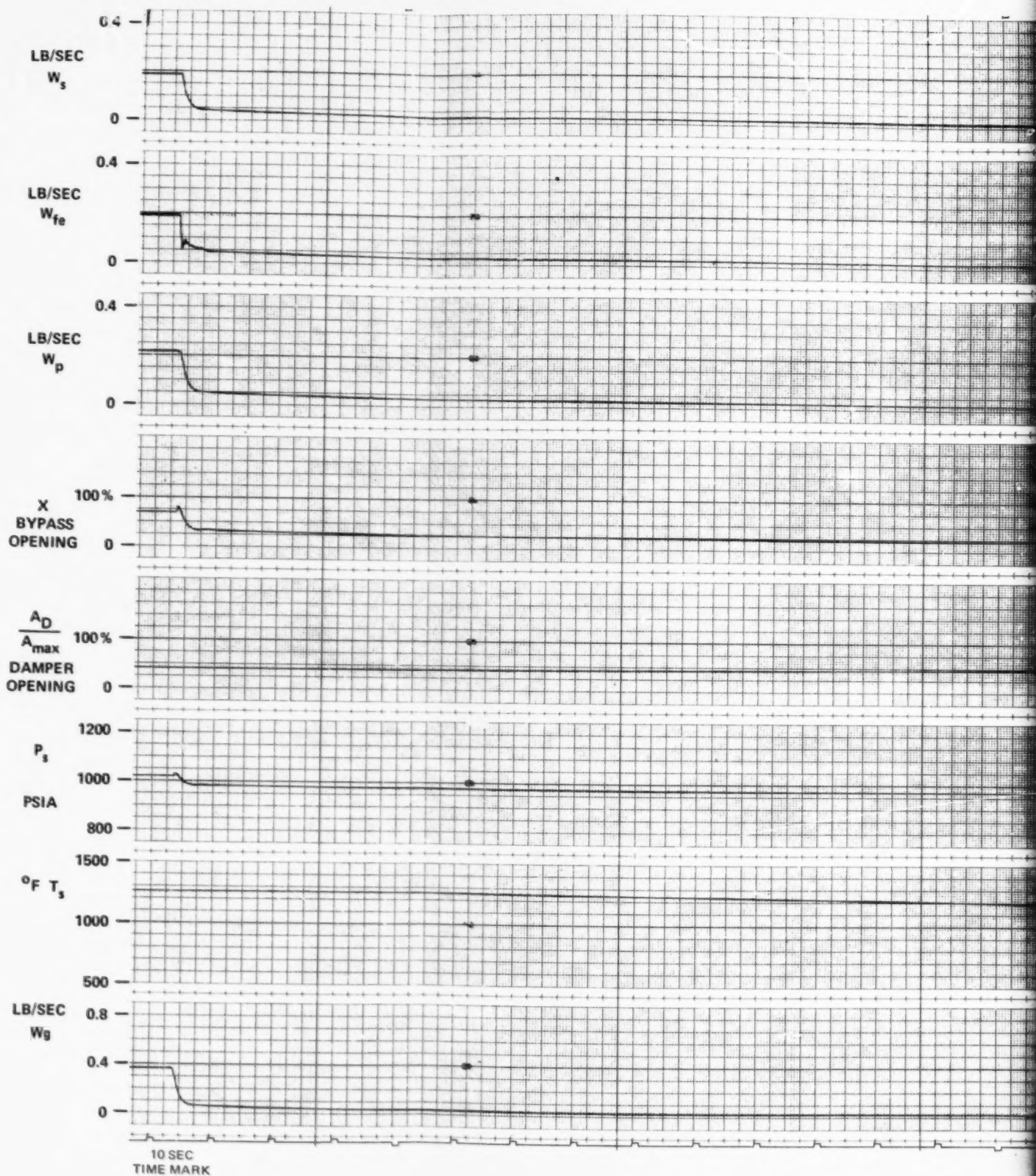
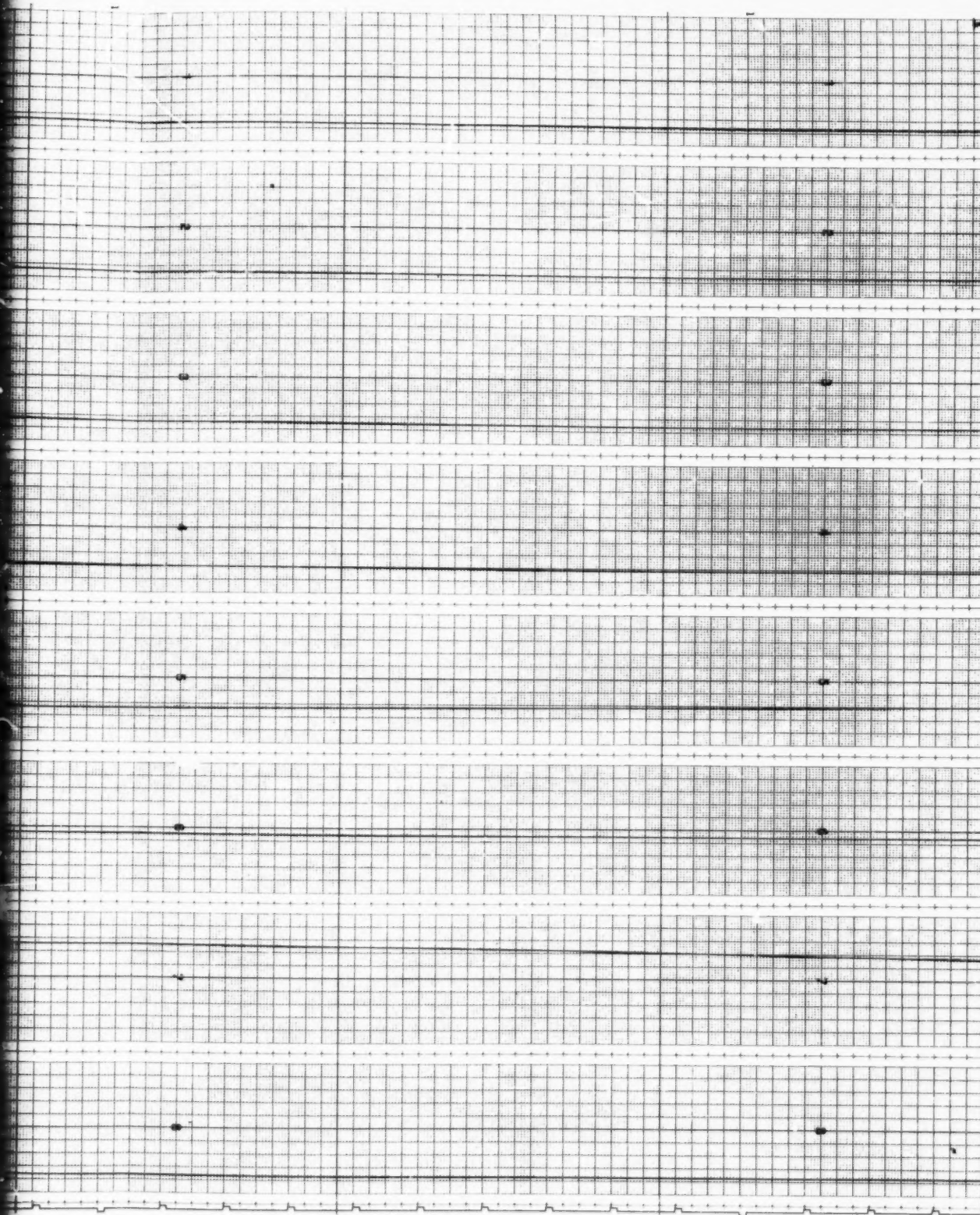
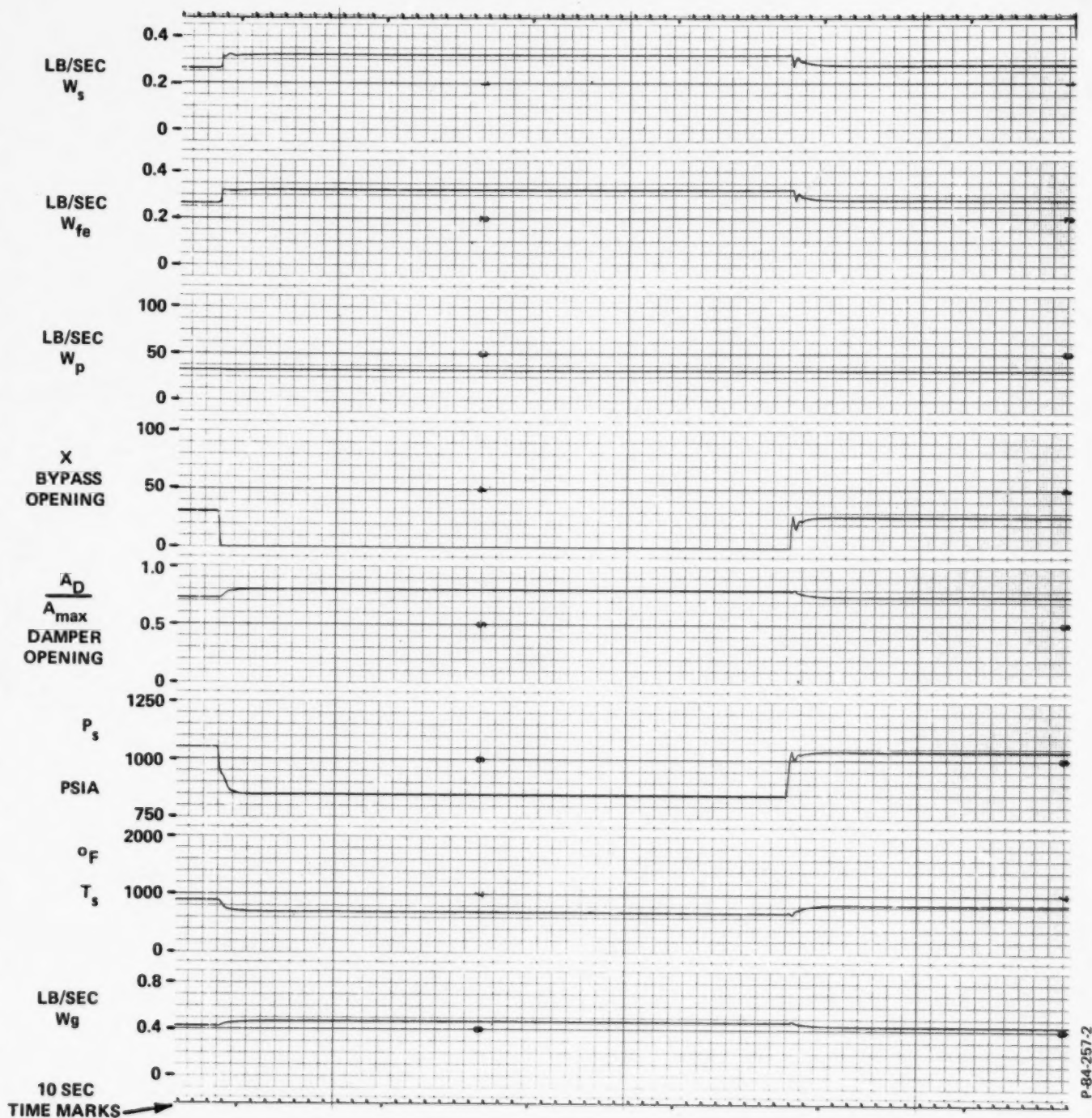


Figure 3-6 - Hybrid Computer Results; Fast Equal Percentage Bypass Valve



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Figure 3-8 - Hybrid Computer Results; Revised Boiler -
Operation Above 100 Percent Capacity

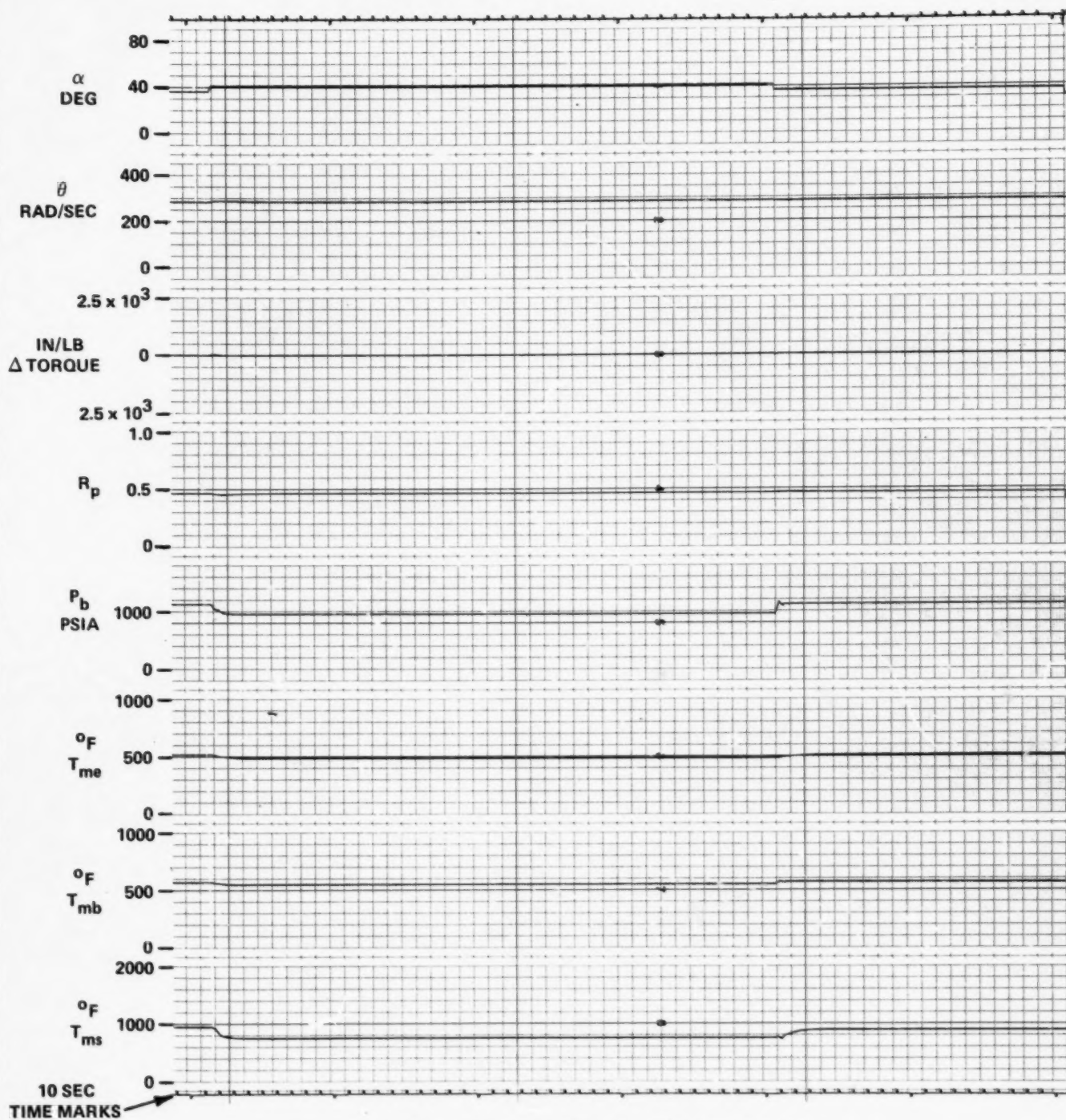
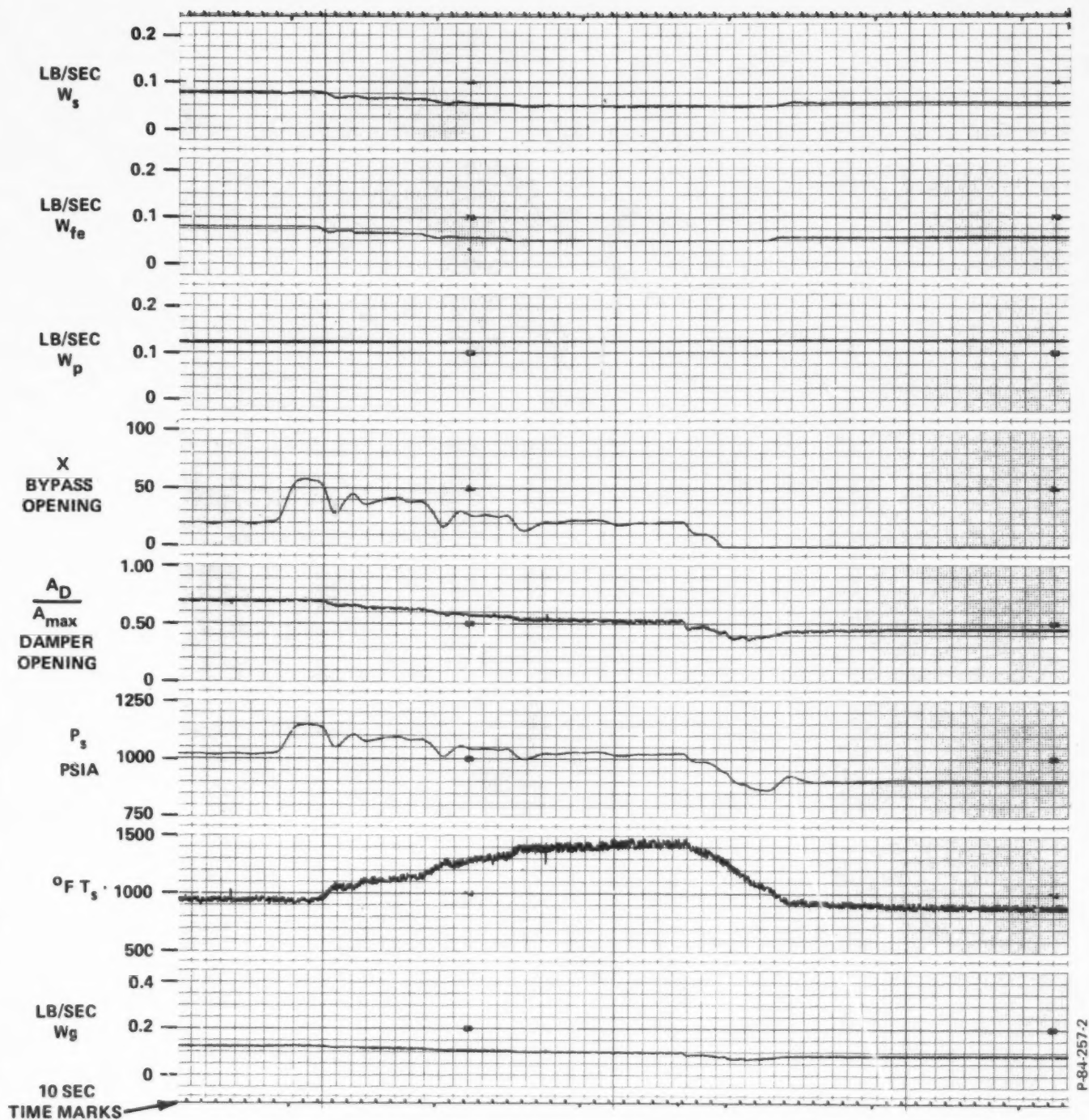


Figure 3-9 - Hybrid Computer Results; Revised Boiler -
Operation Above 100 Percent Capacity

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Figure 3-10 - Hybrid Computer Results; Revised Boiler - Manual Feedwater Control - Low Gain Temperature Control

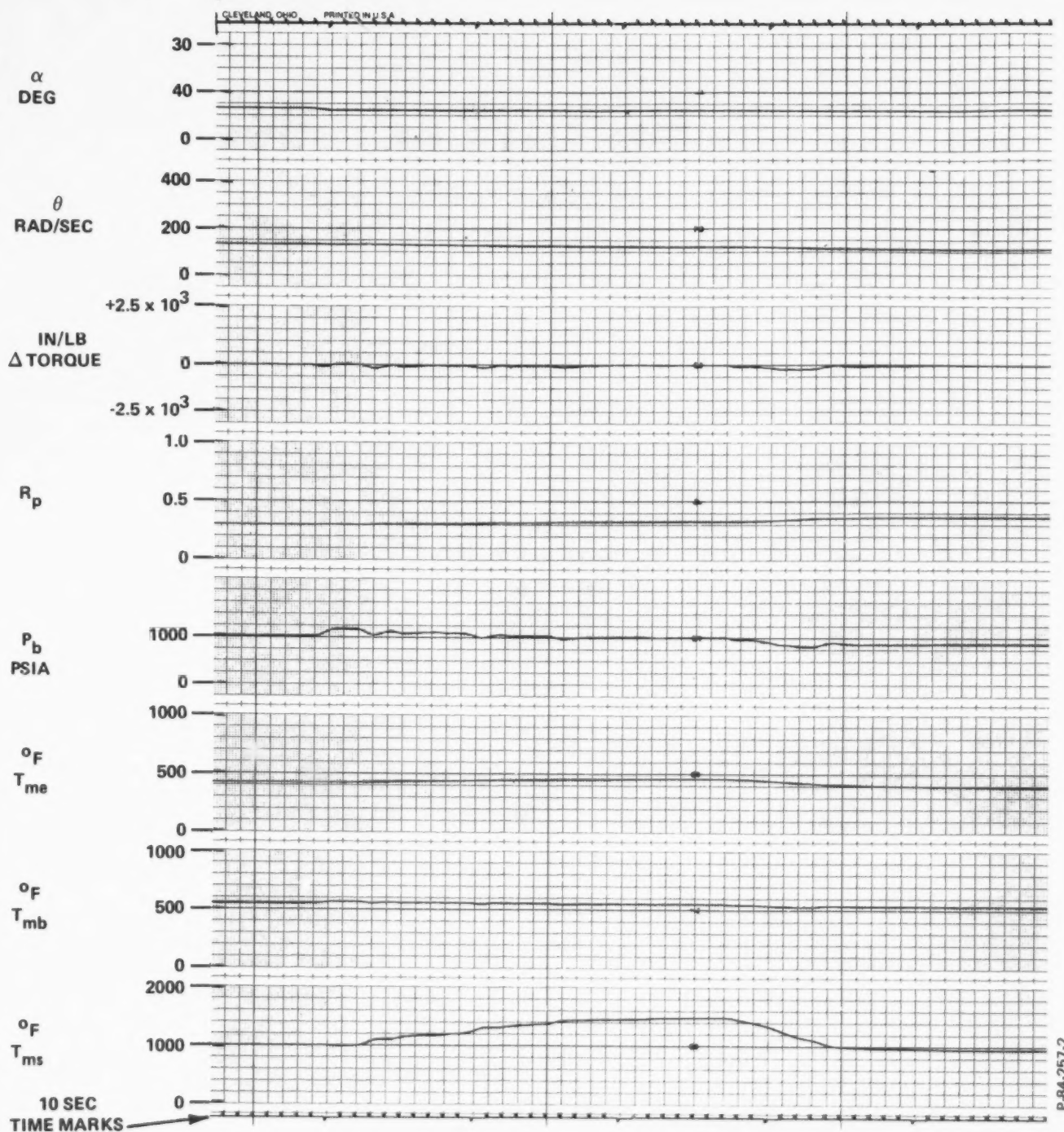


Figure 3-11 - Hybrid Computer Results; Revised Boiler -
Manual Feedwater Control - Low Gain Temperature Control

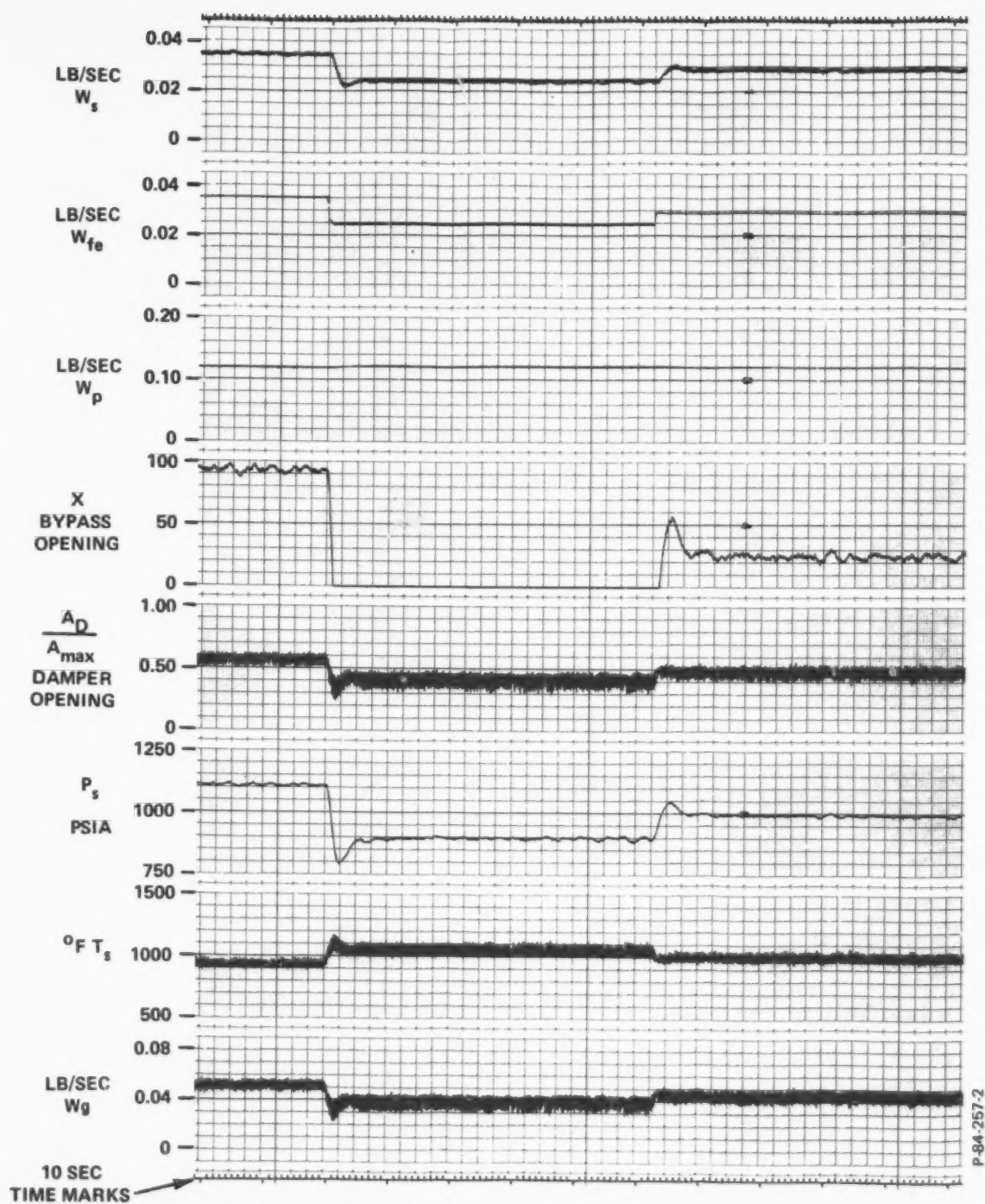


Figure 3-12 - Hybrid Computer Results; Revised Boiler - Manual Feedwater Control - High Gain Temperature Control

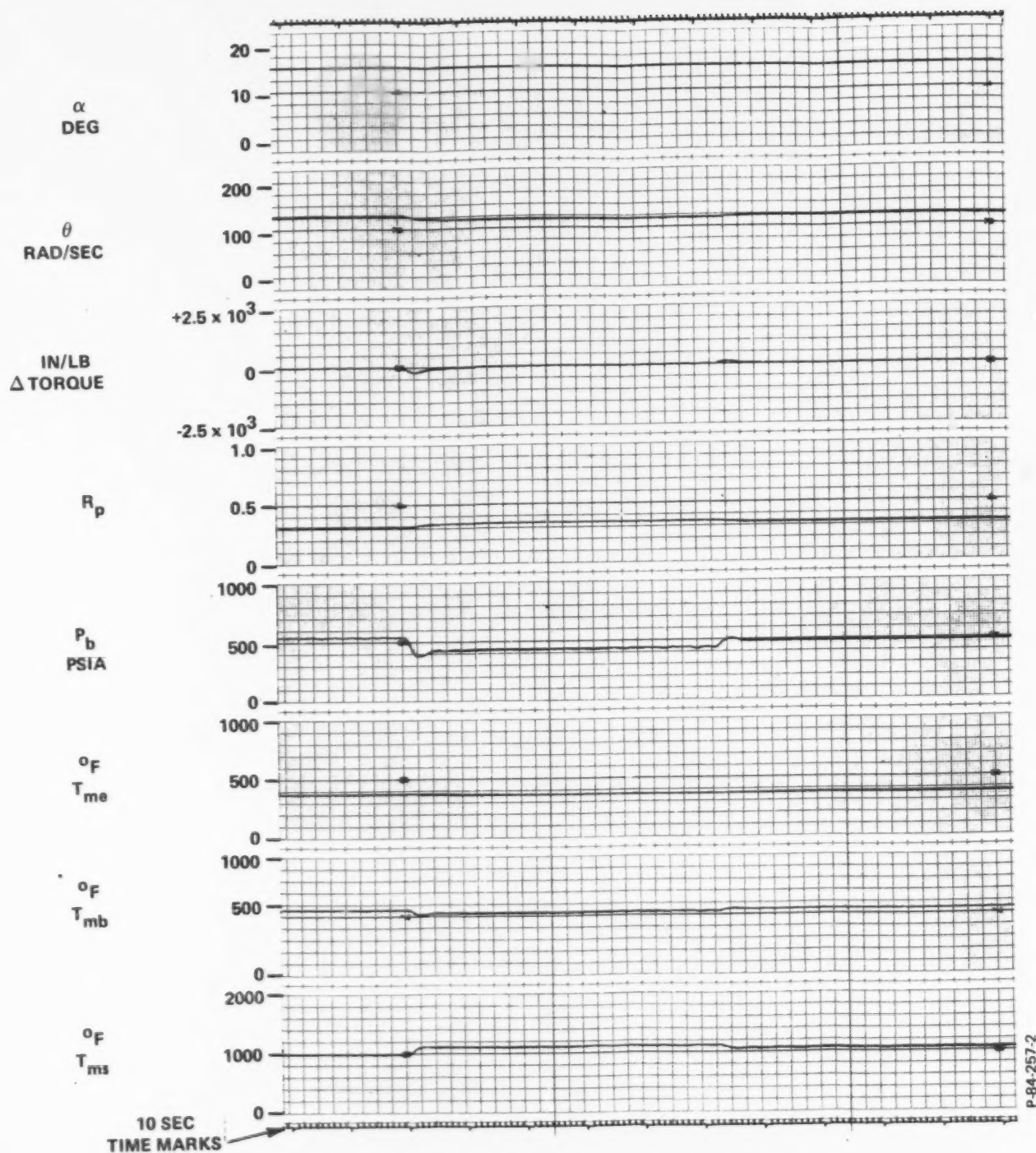


Figure 3-13 - Hybrid Computer Results; Revised Boiler - Manual Feedwater Control - High Gain Temperature Control

SECTION 4

COMPUTER SIMULATION OF THE BURNER SYSTEM

During the previous reporting period a mathematical model of the burner system was devised. This model was used to write a computer program for a time-shared computer with which the effects of blower speed and inlet throttling valve position on combustion gas flow and other burner system parameters were studied. A brief summary of some of the results was reported in the Bendix Research Laboratories Quarterly Report No. 3, December 10, 1971.

During the present reporting period, the mathematical model was reexamined and slightly revised. The computer simulation was also changed to reflect the revisions to the mathematical model. Additional parameter studies were conducted to investigate the effect of the heat exchanger flow exponent. A description of the burner system model, the pertinent equations, the computer simulation, and the additional results are given below.

In the development of models of any system, a first step is the derivation of the system equations. In general, the equations are based, in part, on certain assumptions which are made not only to facilitate the modeling task but also are those assumptions which result in the least sacrifice of accuracy. Some of the basic assumptions made concerning the present system model are:

- All gas flow through restrictors is proportional to the $1/2$ power of the pressure drop except for the exhaust restrictor. Several exponents, including 1, $1/2$, and $1/1.8$, were evaluated for the exhaust restrictor.
- The specific heats of the recirculated exhaust gas (EGR) and the inlet air-EGR mixture are assumed equal to the specific heat of the inlet air.
- The temperature rise in the combustor is 3000°F .
- The air/fuel ratio is constant at 20:1.
- Heat transfer from the burner system to the surroundings is neglected.
- Injected fuel is completely vaporized.

A schematic block diagram of the burner system is given in Figure 4-1. Gage pressures shown in the figure are those for maximum exhaust flow and were used to compute the orifice coefficients for the various restrictors. The symbols used in the block diagram along with the variables and constants used in the equations to be presented are all to be

found in the nomenclature at the end of this section. For each symbol, the nomenclature includes a brief description of the symbol, the units, and, where applicable, the numerical value.

Basically, in the system model, incoming air, WA, at atmospheric temperature and pressure is mixed with a prescheduled amount of recirculated exhaust gas, WR. This gas then passes through the combustion air blower. Fuel, WF, is mixed with the air-exhaust mixture which then passes on to the combustor and is burned, producing a temperature rise, and is exhausted to atmosphere as WE. The system equations are presented below.

1. $WG = WA + WR$
2. $WM = WG + WF$
3. $WF = 0.05 WA$
4. $WM = WE + WR$
5. $WA = \frac{(PAB - PI)^{1/2}}{RI}$
6. $WG = \frac{(PI - PG)^{1/2}}{RG}$
7. $WM = \frac{(PO - PCI)^{1/2}}{RD}$
8. $WE = \frac{(PCO - PAB)^{1/1.8}}{RC}$
9. $WR = \frac{(PCO - PI)^{1/2}}{RR}$
10. $PB = PO = PF$

11. $PCO = PCI$
12. $PB = PG + f(SP, VG)$
13. $TI = \frac{WA \times CPA \times TAB + WR \times CPR \times TCO}{WA \times CPA + WR \times CPR}$
14. $TB = TI$
15. $TO = \frac{WG \times CPG \times TB + WF \times CPF \times TF - HLG \times WF}{WG \times CPG + WF \times CPF}$
16. $TCI = TO$
17. $TCO = TCI + 3000$
18. $VG = WG/D$
19. $D = \frac{PG \times 0.03613 \times 1728}{640 \times (460 + TG)}$
20. $CPA = CPR = CPG$
21. $TG = TI$
22. $TE = TCO$

Equation (12) above contains a function $f(SP, VG)$, and defines blower outlet pressure PB as the sum of the inlet pressure and some relationship between blower speed and volumetric flow rate. This equation was evaluated by means of a three-dimensional data matrix consisting of blower pressure rise, blower speed, and volumetric flow rate. The blower manufacturer's data matrix was entered into the program as input data. Linear interpolation between values in the data matrix was accomplished by means of a subroutine. This method of handling blower characteristics facilitates the modeling of different blowers inasmuch as the

only requirement is the replacing of the data matrix. A complete listing of the simulation program is given at the end of the section.

A simulation of the burner system with two types of inlet throttling schemes was made, one representing the shutter using a triangular opening and the other a simple throttle plate.

The flow area versus position, for a properly sized damper of the construction using the triangular shutter mechanization similar to the air damper used in the control mode test program, is given by

$$A = 1.5397 + 0.01885 \theta^2$$

Also, RG can now be expressed as:

$$RG = \frac{1}{0.0573 + 0.0007 \theta^2}$$

The equation can now be inserted in the program for RG and the effects of varying θ can be determined. The exhaust gas flow variation with blower speed for this throttling scheme is shown in Figure 4-2 for an exponent of 1.0, representing laminar flow through the restriction RC.

Next, assume a butterfly valve having the following characteristics:

$$A = 1.5397 + 46.1926 (1 - \cos \theta)$$

and

$$RG = \frac{1}{[0.0573 + 1.7202 (1 - \cos \theta)]}$$

The effects of this throttling scheme on exhaust gas flow versus blower speed is also shown in Figure 4-2 with the throttle angle given in parentheses. Examination of the results shows outflow to be quite linear with respect to blower speed for a given inlet area over the usable speed range of the blower.

From the results shown in Figure 4-2, a new set of curves may be derived. They are exhaust outflow versus blower inlet valve position, with blower speed as a parameter. These curves are the curves of constant blower speed shown in Figure 4-3. It can be inferred from the

nonlinearity of these curves that some nonlinear response or curve shaping might be required for a shutter or throttle butterfly-valve inlet control.

The effect of changing the flow exponent to $1/1.8$, was evaluated. This figure corresponds more closely to recent SES experimental results of steam generator pressure-flow tests. The calculation results, as far as pressure-flow relations are concerned, remained nearly unchanged as shown in Figure 4-4.

The block diagram indicates that no active EGR valve exists and EGR flow is determined by system pressure distribution. Assuming no heat transfer from the gases, the $1/2$ flow exponent model indicates a constant 218°F combustion inlet temperature. The $1/1.8$ flow exponent model gives a blower inlet temperature of 247°F at minimum inlet flow and 216°F blower inlet temperature at the maximum inlet flow. At the extreme, if laminar flow conditions are assumed in the heat exchanger and a flow exponent of 1.0 is used, the blower inlet temperature varies from 502°F at minimum flow conditions to 212°F at minimum flow condition. The temperature calculation was made with no heat transfer to the surroundings. The effect of heat transfer, however, is believed to be significant and will affect the results. The prime purpose of the present study was the determination of pressure-flow control characteristics for the system studies, and did not require accurate temperature predictions.

NOMENCLATURE

<u>Symbol</u>	<u>Units</u>	<u>Value</u>	<u>Description</u>
CPA	btu/lb°R	0.243	Specific heat of air
CPF	btu/lb°R	0.4	Specific heat of gasoline
CPG	btu/lb°R	---	Specific heat of inlet air and EGR mixture
CFR	btu/lb°R	---	Specific heat of EGR
D	lb/ft ³	---	Density of blower inlet gas
HLG	btu/lb	156	Heat of vaporization of gasoline
PAB	psia	14.7	Atmospheric pressure
PB	in H ₂ O	---	Blower outlet pressure
PCI	in H ₂ O	---	Combustor inlet pressure
PCO	in H ₂ O	---	Combustor outlet pressure
PF	in H ₂ O	---	Fuel pressure
PG	in H ₂ O	---	Blower inlet pressure
PI	in H ₂ O	---	Pressure of inlet air + EGR mixture
PO	in H ₂ O	---	Pressure of PI + fuel mixture
RC	(in) ^{1/1.8} (sec/lb)	2.83	Combustor restriction
RD	$\sqrt{\text{in sec/lb}}$	2.54	Blower + duct outlet restriction
RG	$\sqrt{\text{in sec/lb}}$	1.09	Blower inlet restriction
RI	$\sqrt{\text{in sec/lb}}$	1.96	Inlet restriction
RR	$\sqrt{\text{in sec/lb}}$	68.4	EGR duct restriction
SP	rev/min	---	Blower speed
TAB	°F	70	Inlet air temperature

<u>Symbol</u>	<u>Units</u>	<u>Value</u>	<u>Description</u>
TB	°F	---	Blower outlet temperature
TCI	°F	---	Combustor inlet temperature
TCO	°F	---	Combustor outlet temperature
TG	°F	---	Blower inlet temperature
TI	°F	---	Temperature of inlet air + EGR mixture
TO	°F	---	Temperature of TI + fuel mixture
WA	lb/sec	---	Inlet air flow
WE	lb/sec	---	Exhaust gas flow
WF	lb/sec	---	Fuel
WG	lb/sec	---	Inlet air + EGR flow
WM	lb/sec	---	WG + fuel mixture flow
WR	lb/sec	---	EGR flow
VG	ft ³ /min	---	Blower volumetric flow rate

Computer Program Listing (1 of 3)

GASCK

```

100 COMMON DA(14),DB(15),DC(14,15)
110 READ,(DA(I),I=1,14)
120 READ,(DB(I),I=1,15)
130 READ,((DC(I,J),J=1,15),I=1,14)
140 READ,TAB,PAB,TF
150 READ,CPA,CPF,HLG
160 READ,RI,RD,RC,RK
162 ARC=1.8
164 RC=3**((1/XRC)
165 RC=RC/.65
166 PRINT"RC- ",RC
170 THE1=35.
180 122 CONTINUE
190 PRINT,(*,I=1,7)
200 CPR=CPA
210 CPG=CPA
220 AREA=1.5397+.018854*(THE1)**2.
230 RG=1./(.03724*AREA)
240 THE2=1.-((AREA-1.5397)/46.1926)
250 CPR=CPA
260 PRINT,(*,I=1,5)
270 PRINT,RG,THE1,AREA
280 PRINT,THE2
290 PRINT
300 WA=.05
310 50 WR=((1.05*RC*WA)**XRC+RI*RI*WA*WA)**.5)/RK
320 PWR=WR/WA*100.
330 WG=WA+WR
340 WF=.05*WA
350 WM=WF+WG
360 WE=WM-WR
370 A=WG*CPG+WF*CPF
380 TB=(WA*CPA*TAB*A+WR*CPR*(WF*CPF*TF-HLG*WF+3000.*A))/(A*(
390 +WA*CPA+WR*CPR)-WR*CPR*WG*CPG)
400 PG=PAB-(RI*WA)**2.-(RG*WG)**2.
410 PB=PAB+(RC*WE)**XRC+(RD*WM)**2
420 B=PB-PG
430 TJ=(WG*CPG*TB+WF*CPF*TF-HLG*WF)/A
440 TCJ=TJ+3000.
450 D=(.03613*1728.*PG)/(640.*(460.+TG))
460 C=WG*60./D
470 IF(11.-B)322,312,312
480 312 CONTINUE
490 CALL INPOL(B,C,SP)
500 GJ TJ 329
510 322 SP=0.
520 GJ TJ 329
530 329 CONTINUE
540 PRINT,WA,WG,WM,WE,WR,PWR,TB,TCJ,PG,PB,B,SP,C,D
550 PRINT

```

Computer Program Listing (2 of 3)

GASCK CONTINUED

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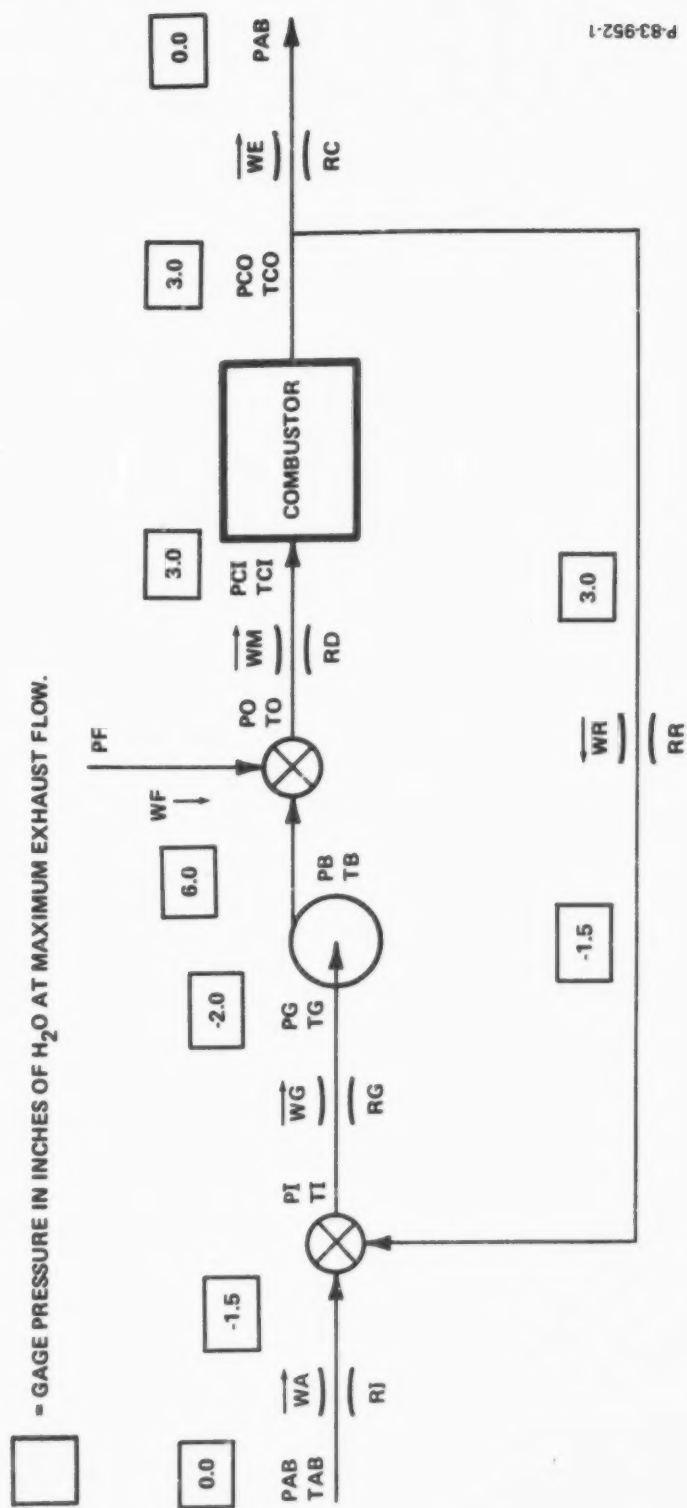
560 WA=WA+.05
570 IF(.7-WA)230,230,50
580 230 CONTINUE
590 IF(0.-THE1)372,379,379
600 372 THE1=THE1-7.
610 G0 T0 122
620 379 CONTINUE
630 END
640 SUBROUTINE INPOL(XA,XB,XC)
650 COMMON AA(14),BB(15),CC(14,15)
660 I=1
670 40 IF(XA-AA(I))70,50,50
680 50 I=I+1
690 G0 T0 40
700 70 M1=I-1
710 M2=I
720 I=1
730 100 IF(XB-BB(I))130,110,110
740 110 I=I+1
750 G0 T0 100
760 130 N1=I-1
770 N2=I
780 GG=(XB-BB(N1))/(BB(N2)-BB(N1))
790 CL0=CC(M1,N1)+(CC(M1,N2)-CC(M1,N1))*GG
800 CHI=CC(M2,N1)+(CC(M2,N2)-CC(M2,N1))*GG
810 XC=CL0+(CHI-CL0)*((XA-AA(M1))/(AA(M2)-AA(M1)))
820 RETURN
830 END
840 $DATA
850 0.,.5,1.,1.5,2.,3.,4.,5.,6.,7.,8.,9.,10.,11.
860 0.,.50.,100.,150.,200.,250.,300.,350.,400.,450.,500.,
870 550.,600.,650.,700.
880 0.,325.,651.,976.,1302.,1627.,1953.,2278.,
890 2604.,2929.,3255.,3580.,3906.,4231.,4557.,
900 1043.,1075.,1189.,1375.,1603.,1882.,2202.,2527.,
910 2856.,3188.,3522.,3858.,4194.,4532.,4871.,
920 1489.,1514.,1564.,1710.,1894.,2108.,2351.,2640.,
930 2958.,3280.,3606.,3935.,4266.,4599.,4933.,
940 1853.,1854.,1876.,1986.,2152.,2341.,2553.,2793.,
950 3053.,3368.,3687.,4010.,4336.,4664.,4994.,
960 2125.,2143.,2151.,2232.,2379.,2553.,2749.,2965.,
970 3205.,3452.,3765.,4082.,4403.,4727.,5054.,
980 2617.,2627.,2624.,2662.,2772.,2930.,3104.,3296.,
990 3502.,3733.,3975.,4223.,4533.,4849.,5165.,
1000 3034.,3036.,3027.,3050.,3129.,3259.,3420.,3597.,
1010 3789.,3994.,4217.,4457.,4702.,4966.,5230.,
1020 3402.,3396.,3383.,3392.,3444.,3559.,3709.,3875.,
1030 4053.,4247.,4453.,4672.,4911.,5150.,5389.,
1040 3736.,3723.,3708.,3712.,3753.,3843.,3972.,4132.,
1050 4303.,4483.,4681.,4887.,5106.,5325.,5544.,

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Computer Program Listing (3 of 3)

GASCK CONTINUED

1060 4044., 4024., 4007., 4007., 4037., 4105., 4221., 4373.,
1070 4536., 4712., 4895., 5097., 5299., 5501., 5703.,
1080 4331., 4304., 4286., 4283., 4301., 4353., 4464., 4593.,
1090 4757., 4927., 5105., 5283., 5461., 5639., 5817.,
1100 4587., 4567., 4547., 4541., 4550., 4601., 4693., 4811.,
1110 4968., 5125., 5282., 5439., 5596., 5753., 5910.,
1120 4835., 4816., 4795., 4786., 4792., 4836., 4910., 5025.,
1130 5153., 5281., 5409., 5537., 5665., 5793., 5921.,
1140 5076., 5053., 5030., 5018., 5025., 5060., 5166., 5272.,
1150 5378., 5484., 5590., 5696., 5802., 5908., 6014.,
1160 70., 407., 70.,
1170 .243., .4, 156.,
1180 1.96, 2.54, 4.62, 68.4



P-83-952-1

Figure 4-1 - Schematic Block Diagram of Burner System

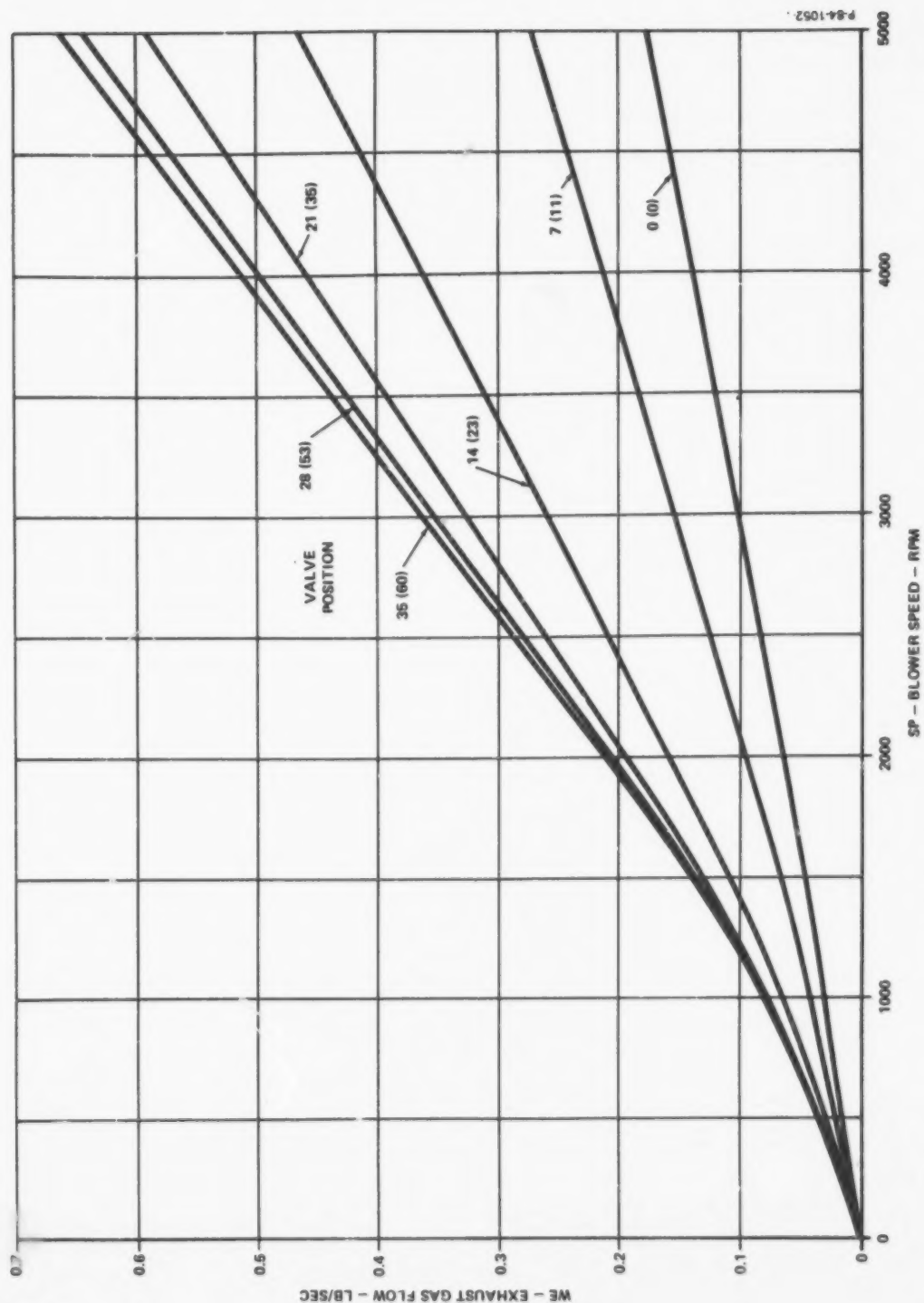


Figure 4-2 - Exhaust Gas Outflow versus Blower Speed

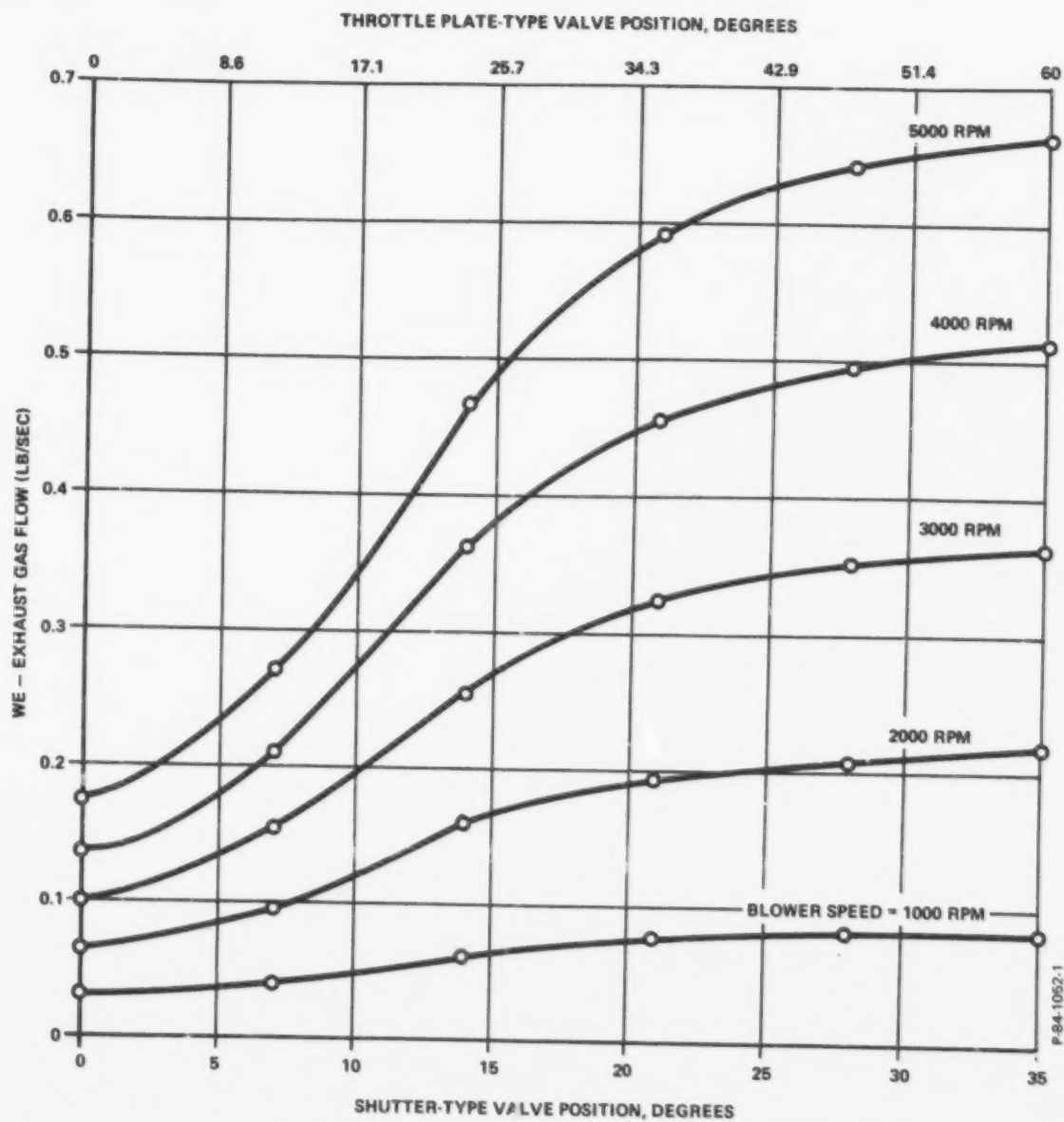


Figure 4-3 - Exhaust Gas Outflow versus Blower Inlet Valve Position - Flow Exponent = 1.0

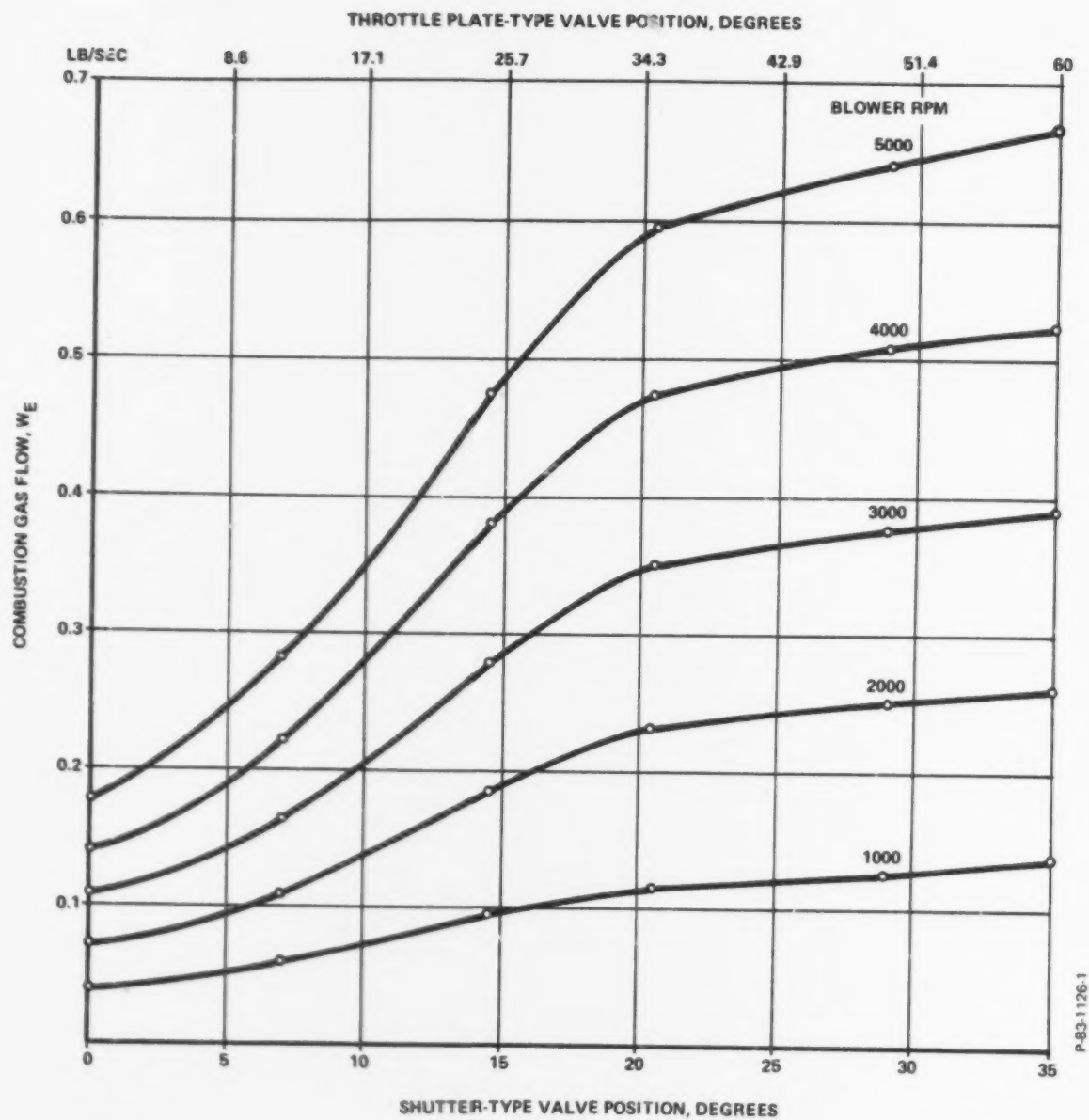


Figure 4-4 - Exhaust Gas Outflow versus Blower Inlet Valve Position - Flow Exponent = 1.8

SECTION 5

FEEDWATER CONTROL

5.1 DIRECT PRESSURE OPERATED BYPASS VALVE DESIGN

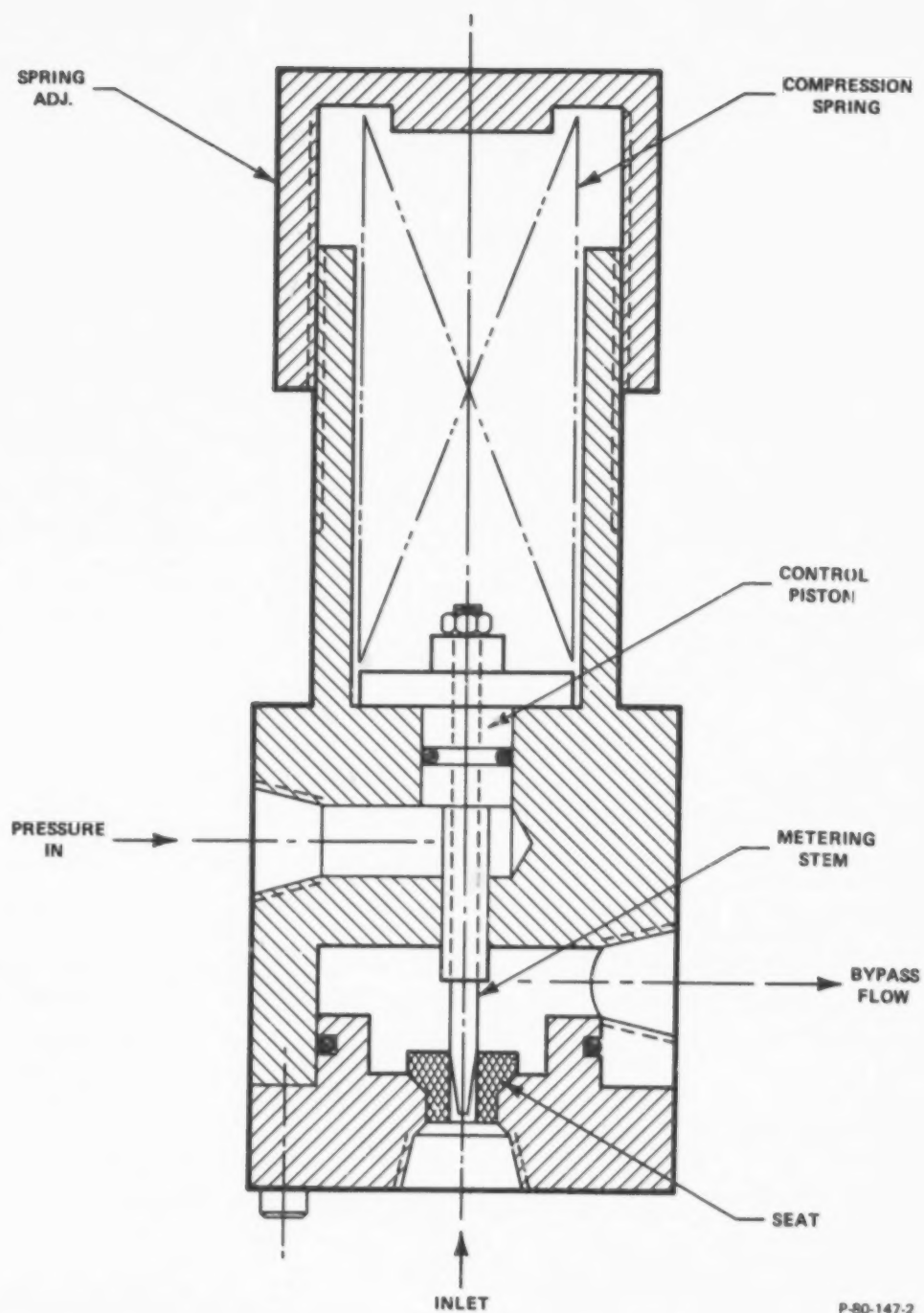
During the reporting period a concept of feedwater control, wherein bypass flow would be a direct function of steam generator outlet pressure alone, was investigated. A preliminary valve actuator design was made to determine valve and valve actuator parameters such as orifice size, spring rate, moving mass, and actuator piston area. These parameters were incorporated into the hybrid computer simulation and the effects of this control method were studied. The results obtained indicated that further development of this concept would be fruitful. The preliminary design is shown pictorially in Figure 5-1.

The preliminary design was subsequently modified to permit the possibility of installing various piston and spring combinations and also to accommodate an external actuator should circumstances so indicate. Referring again to Figure 5-1, the operation of the valve is as follows. Pressure on the effective area of the control piston causes the piston to move upward when the force on the piston exceeds the compression spring preload, thus extracting the metering stem from the seal and increasing the bypass flow. The spring and piston area have been sized so the valve will crack at 950 psi and flow 3.0 gpm at approximately 1250 psi. A stellite metering stem and seat combination has been obtained. Typical lift flow curves of metering stem and seat combinations available are shown in Figure 5-2. The combination obtained for the application is of the equal percentage type. It may be seen from the figure that the lift flow characteristic of the combination chosen is such that the effective flow area is approximately proportionate to the square of the lift.

5.2 FABRICATION AND TESTING OF THE BYPASS VALVE

The completed bypass valve design has been released for fabrication and completion is anticipated during the next reporting period. All purchased items have been ordered and most items have been received.

After completion of fabrication and assembly, the bypass valve will be given preliminary tests to assure proper valve opening and closing, correct pressure calibration, and freedom from leaks. After proper bypass valve operation is ascertained, the valve will be incorporated into the test setup shown schematically in Figure 5-3. Detailed test plans will be formulated and executed during the next reporting period.



P-80-147-2

Figure 5-1 - Single Stage Bypass Valve Schematic

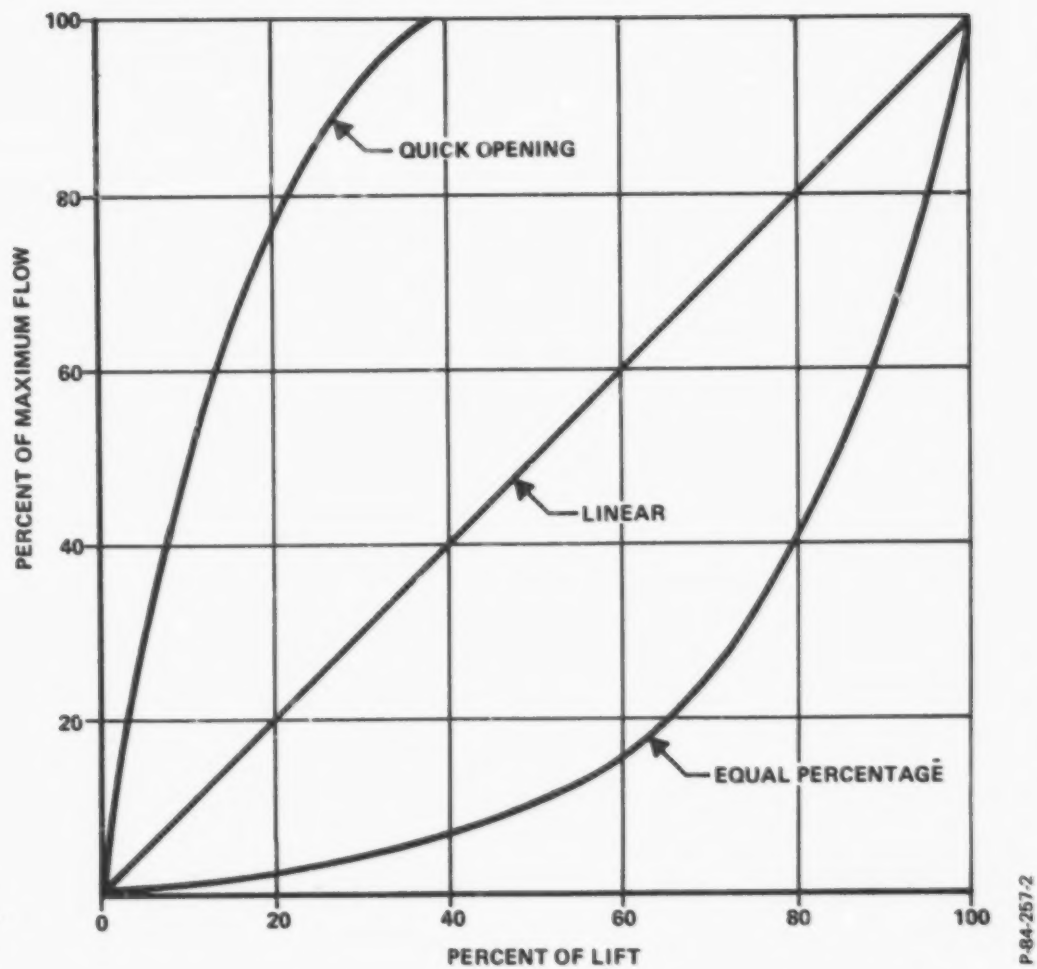


Figure 5-2 - Typical Lift Flow Curves for Research Control Valves

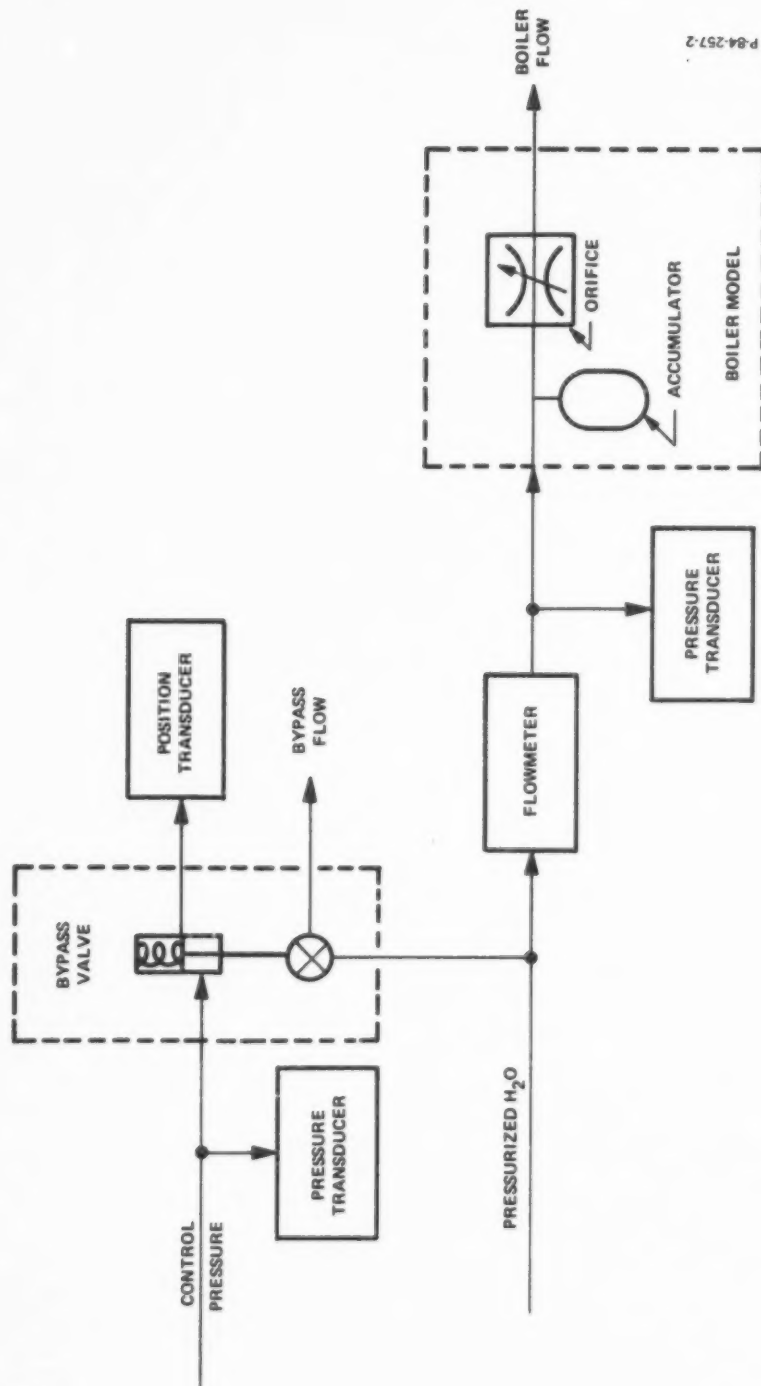


Figure 5-3 - Test Set-Up Schematic for Feedwater Control

SECTION 6

FUEL/AIR RATIO CONTROL

6.1 FUEL SCHEDULING

During the reporting period, preliminary fuel scheduling requirements, as determined by S.E.S., were transmitted to the Bendix Research Laboratories. These requirements are shown in Figure 6-1 as a graph of air/fuel ratio as a function of inlet air flow in lb/hr. The stoichiometric ratio has been taken as 14.7, and therefore the maximum ordinate (22.05) represents 50% excess air. The maximum or 100% inlet air flow has been taken as 2000 lb/hr. The maximum fuel flow rate is therefore 113.38 lb/hr and the total gas flow to the burner is then 2113.38 lb/hr.

6.2 RSA UNIT FABRICATION

A Bendix RSA-10AAl fuel control device has been placed on order and is presently being fabricated at Energy Controls Division in South Bend, Ind. Modifications to the basic unit are being incorporated so that the unit performance will conform as closely as possible to the system requirements, and be capable of tracking the desired fuel schedule with sufficient accuracy. In addition to the basic features described in reports and pertinent literature previously submitted, the following features are being incorporated in the unit presently being fabricated:

- Boost venturi - this feature will enhance the accuracy of the unit operation at low inlet air flows and provide as low a pressure drop as possible across the unit.
- Idle enrichment valve - this feature is provided to improve the control of the air/fuel mixture near the cut-off point, and improve the ability of the unit to accurately track the desired fuel schedule curve.

A manual mixture control and idle cut-off lever will be provided so that deviations from the desired fuel schedule curve can be corrected, or additional schedules may be tried. An automatic mixture control, which would provide a measure of inlet air density compensation, will not be provided at this time. Fabrication of the unit is expected to be completed during the next reporting period.

A functional block diagram of the RSA fuel control system is shown in Figure 6-2. Briefly, the system operates as follows:

- (1) Pressurized fuel from the fuel pump flows through the manual mixture control and idle cut-off valve to one side of both the enrichment valve and fuel diaphragms, and also to the enrichment and cruise jets.

- (2) If the sum of the spring force and the pressure of the fuel from the enrichment jet exceeds the fuel pressure from the manual mixture control valve, the enrichment diaphragm moves to open the enrichment valve. Thus in the power range (manual idle valve open greater than idle setting) fuel may flow through the series combination of the fixed enrichment jet and variable enrichment valve and in parallel with the flow through the fixed cruise jet.
- (3) Air is drawn into the unit through the inlet and boost venturi which causes a pressure drop. The differential pressure across the venturi is applied to either side of the air diaphragm. Thus the net force on the air diaphragm is a function of the air flow through the unit.
- (4) Fuel leaving the manual idle valve is applied to the remaining side of the fuel diaphragm and the variable ball valve. The air and fuel diaphragms are constructed so that the fuel diaphragm opposes the force of the air diaphragm. Since the net force on the air diaphragm is totally dependent on air flow, and since the pressure on one side of the fuel diaphragm is fixed at any point in time, the two diaphragms will balance only when the remaining fuel pressure, that to and through the ball valve, is of the proper value, thus ensuring that fuel flow is a function of air flow.

From the above, it should be evident that the fuel control system is a closed-loop system with input air flow as an input control parameter. It should further be evident that by varying the sizes and taper of the enrichment jet and the spring which bears against the enrichment valve control diaphragm, the unit can be tailored to track nearly any fuel schedule.

6.3 TEST PLANS FOR BENDIX RSA FUEL CONTROL DEVICE

Upon completion of the mechanical fabrication of the RSA unit, the unit will be calibrated at the 25% (500 lb/hr, 22.05 A/F ratio) and the 100% (2000 lb/hr, 17.64 A/F ratio) inlet air flow points. The unit will then be operated over the fuel schedule curve and adjusted for maximum tracking accuracy.

Under the supervision of Bendix Research Laboratories personnel, the following tests will be performed:

- Air flow versus unit pressure drop
- Fuel schedule tracking
- Fuel flow versus fuel pressure with air flow as a parameter

It is presently anticipated that all of the aforementioned calibrations and tests will be performed at Energy Controls Division. After satisfactory performance of the unit is ascertained, the unit will be delivered

to the Bendix Research Laboratories where additional tests may be performed if circumstances so indicate. Ultimate delivery of the unit to S.E.S. will be contingent on future clarification and definition of the interface requirements.

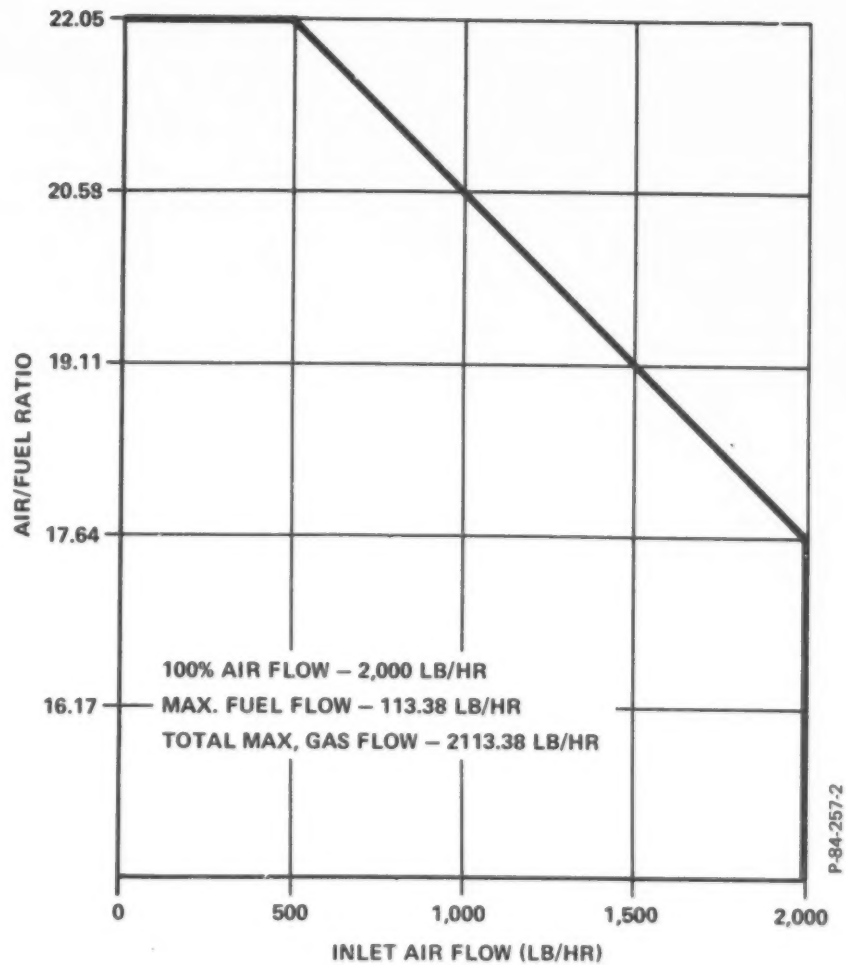


Figure 6-1 - Fuel Schedule - Air/Fuel Ratio Versus Inlet Air Flow

